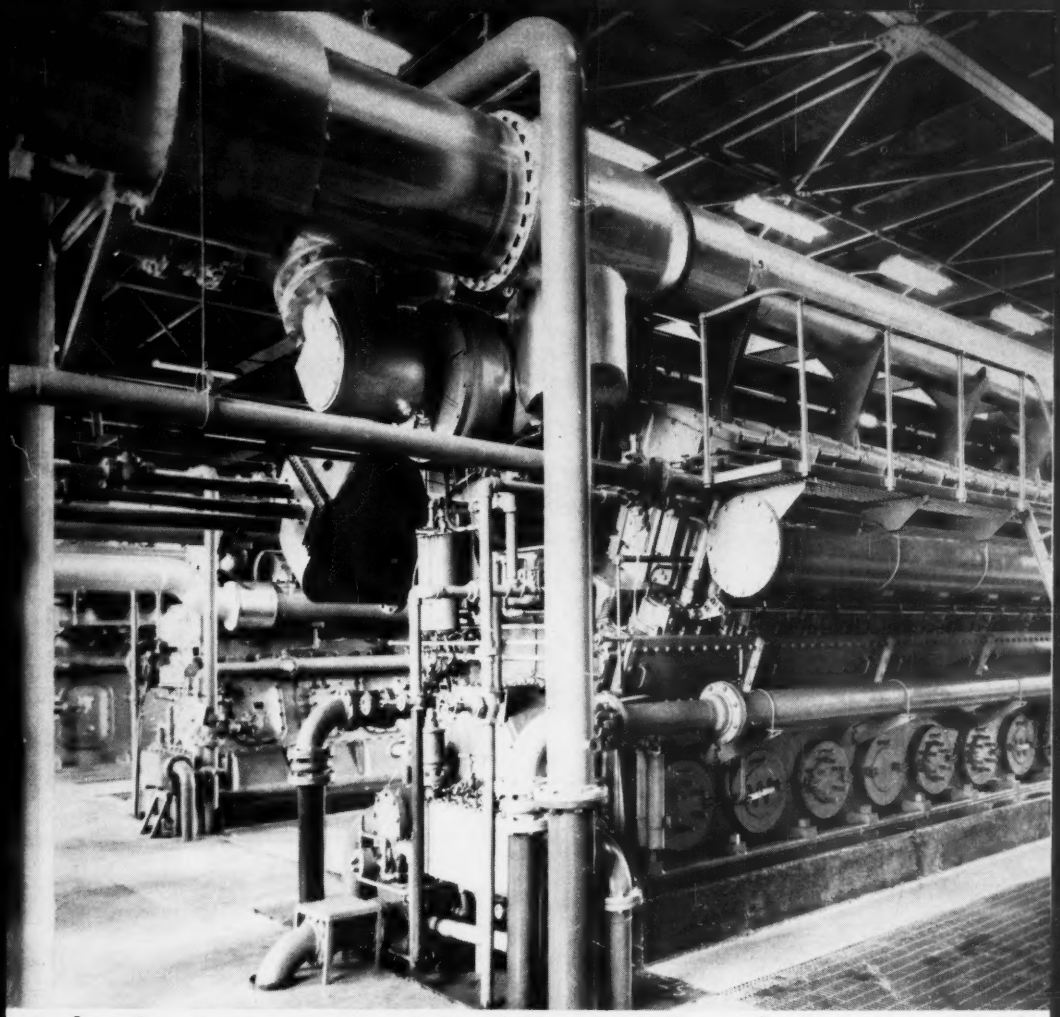
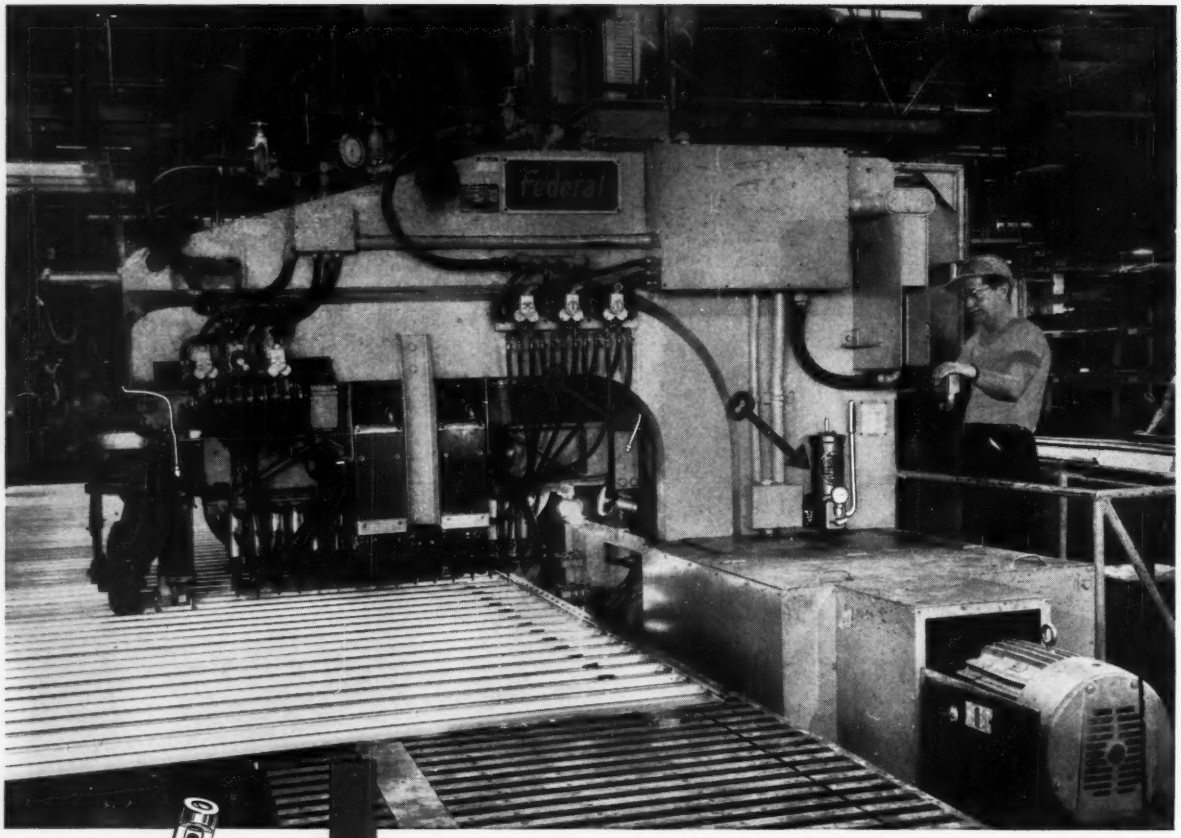
LUBRICATION ENGINEERING

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE



Lube Savings at Municipal Power Plant (see page 426)

JOURNAL OF THE AMERICAN SOCIETY OF LUBRICATION ENGINEERS



FARVAL—Studies in Centralized Lubrication No. 230

Farval lubrication system protects bearings on Federal multi-gun welder

Resistance welders such as this multi-gun Federal, owe much of their effectiveness to their ability to adjust vertical movement of electrodes within exacting limits. Chosen to guard the essential bearings which give this Federal welder its precise control is a modern Farval lubrication system.

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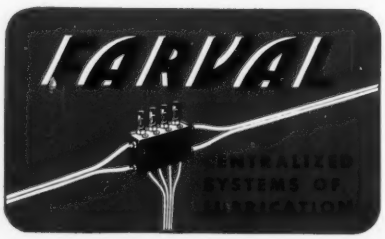
KEYS TO ADEQUATE LUBRICATION

Wherever you see the sign of Farval —familiar valve manifolds, dual lubricant lines and central pumping station—you know a machine is being properly lubricated.



Volume 14, No. 10
October, 1958

Lubrication Engineering

DESIGN RESEARCH APPLICATION EDUCATION MAINTENANCE

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Machinery Sales On Upgrade

Economists are watching each sales-chart jiggle in the machinery industry because that is the place where the first signs of any new upsurge in capital spending are expected to appear. To get a line on how the sales curve will go, *Business Week* surveyed dozens of machinery companies this week and found that "mounting orders have changed glumness to a temperate optimism, often supported by the hard facts of increased orders.

"Of the five major groups of the industry, the makers of pumps and compressors are leading the upward turn. Behind them come the makers of motors and generators, of mining machinery and of machine tools. Even in last place —oil field machinery and tools—there is little discouragement."

The magazine says that "some manufacturers can be pressed into admitting that they're whistling in the dark." But others are standing by their hopes and guesses to the extent of doing some ordering of their own—for steel. Chicago area steelmen say orders from machinery makers are the highest in more than a year.

Group by group, here's what the machinery makers see:

Machine Tools. Orders fell from $24.7 million in June to $22.2 million in July, but many companies see big orders from Detroit once the auto labor contracts are settled. A Cleveland tool maker, typical of many companies, reports orders have risen "gradually" in the past 45 days; he found "the rise nothing startling, but it means the prospects are brightening."

Oilfield Equipment. Orders are up, if only slightly. A Houston company finds July orders a fraction above June, August up 5%—but it adds that "June was our lowest since 1953." Reduced oil production—hit by imports, and by heating conversions to gas—troubles the suppliers. Says one: "There are 700 fewer rigs in operation than a year ago. You'd have holes in your head to buy new rigs with idle ones all around."

Mining Machinery. Companies that mine and prepare coal may spend no more than $150 million on capital equipment this year; the normal rate ranges from $200 million to $250 million. Still, a fourth-quarter pickup is expected, if coal production rises as some signs already indicate. Roberts & Schaefer, a supplier of preparation equipment, says it's getting contracts that have hung for months.

Motors and Generators. Consensus in this group is that orders will continue to rise, though with no boom for some time. Most customers have been "playing it close to the vest." What rise has been felt is unevenly divided. A major manufacturer of medium-weight motors, with fairly steady orders for the past three months, thinks by 1959 there will be a rise of six to seven percent. But the shrunken utility demand for heavy motor and generators is not expected to end before next year.

Pumps and Compressors: This sector sees by far the brightest prospects. A California company says orders are "way out ahead of last year—already 15% over total 1957 sales." An Ohio company says "business is brisk and a backlog of orders is developing." More cautious is the Philadelphia maker of chemical pumps who admits to a pickup, but adds it's "erratic, not steady."

Lube Program To Highlight NLGI Meeting

"Lubricating Grease in Industry" is the theme of the National Lubricating Grease Institute's 26th annual meeting which will conclude NLGI's 25th anniversary year celebration with a program already being promoted as the most thorough and well rounded in the history of the Institute. The oil industry's experts on lubrication will gather in Chicago during October 27-29, at the Edgewater Beach hotel.

Highlighting the theme of the meeting, a special joint session will be held with the American Gear Manufacturer's Association to explore industrial gear lubrication. For research men, the Institute will offer a symposium on rheology of lubricating greases, which will be the most comprehensive on this subject ever presented by any organization to date.

Resins and Putties Safeguard Pumps

Highly corrosion-resistant resins and putties, developed for aircraft and missile applications, are now being experimentally used by a San Diego (Calif.) utility company to cut maintenance costs on sea water-buffeted pumps. The possible savings from the successful control of this maintenance hazard could amount to thousands of dollars annually.

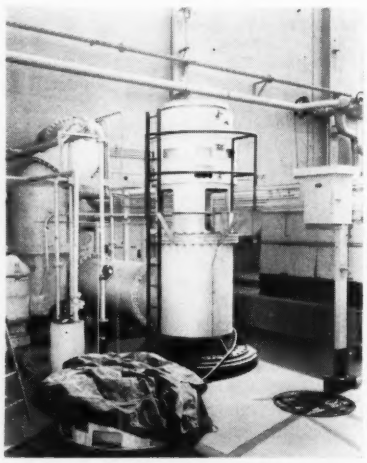
The new application of modern aircraft protective resins is taking place at the Silvergate Power Plant of the San Diego Gas & Electric Company. Here eight Worthington vertical circulating pumps propel 176,000 gallons of sea water a minute through massive steam turbine condensers. The big pumps, working virtually around the clock, present a challenging maintenance problem because of the corrosive punishment of sea water plus the battering action of sand and shell debris.

"This continuous abrasive and corrosive attack," a company spokesman points out, "seriously serrates diffuser vanes and scores deep pits and channels in the diffuser shell which eventually could wear through."

Some months ago, the utility began searching for a protective coating that would solve the costly maintenance problems. Experiments were conducted with a wide variety of protective coatings, including spray metals, welding, rubber compounds, neoprene, ceramics and many others. None solved the problem, and one, peeling from the interior surfaces of pumps, forced a pump shut down.

The utility firm then sought the assistance of an aircraft materials producer, Narmco Resins & Coatings Co. of Costa Mesa, Calif., which recommended the use of one of its putties to fill the pits and channels in diffuser vanes and housing. A liquid resin was recommended to protect the built-up areas against further sea water corrosion and abrasion.

Both products are used extensively in the fabrication of primary aircraft and missile components which are subjected to severe environmental hazards.



One of eight huge, vertical circulating pumps which are in continuous operation at the Silvergate Plant of the San Diego Gas & Electric Company. These pumps gulp in 176,000 gallons of sea water each minute for the condensation of turbine steam.

NEW EMOLEIN® 2910 LOWERS COST OF SYNTHETIC LUBRICANTS

Emolein 2910 Lubricant Ester, formerly designated Emery 3033-S, is the answer to lower-cost synthetic fluids and greases required by the jet age. In synthetic fluids, total cost can be reduced substantially by blending Emolein 2910 with an azelate diester without sacrificing performance. In greases, Emolein 2910 can be used alone or by blending with other fluids.

Emolein 2910 dipropylene glycol dipelargonate is a new type of diester based on inexpensive pelargonic acid. With performance testing having confirmed its utility, it joins the established members of the Emolein line of diesters, Emolein 2957 di-iso-octyl azelate and Emolein 2958 di-octyl azelate, whose performance under the conditions encountered in modern jet engines has given them a recognized position in the synthetic lubricant field.

One more important fact: The Emolein esters are *not* dependent upon imported raw materials. Since they are derived from abundant domestic tallow their future supply is practically unlimited. For more detailed information on Emolein 2910 or the Emolein azelates, return the coupon below.

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"A Survey of Journal Bearing Literature"

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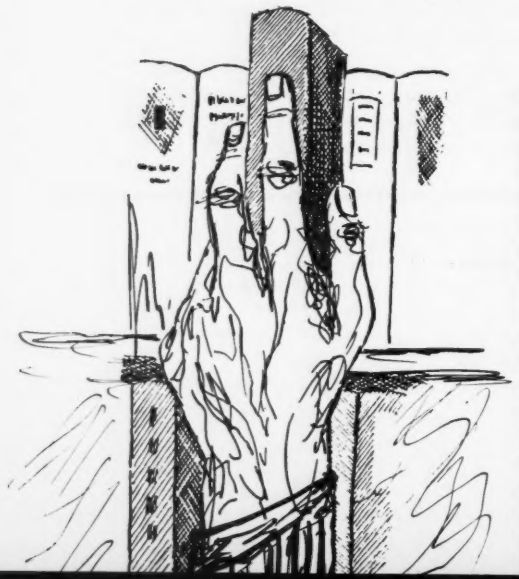

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DISCOVERED!

...new way to keep cutting fluids as fresh as a daisy!

SEE NEXT TWO PAGES

This is one of the causes of emulsion trouble. These are *Pseudomonads*, one of the many harmful types of bacteria found in oil-water emulsions. These, and other bacteria, enter emulsions through the air, water, and plant debris. They set the stage for a chain reaction which causes odor, corrosion, and costly emulsion breakdown.

Prior to Elcide 75, no single inhibitor was able to control the full range of bacterial growth. Certain of these bacteria quickly develop resistance to single inhibitors and continue to flourish. Now, Elcide 75 controls a wider range of bacteria because it is an effective combination of bactericidal agents. This has been proved in shop tests.

In certain areas where waste-oil disposal is a problem, the use of Elcide 75 proportionately reduces the size of the problem.

Elcide 75 also contributes to better products and longer machine-tool life. Bacterial contamination causes acidic decomposition of the emulsion and sets up conditions for corrosion. By controlling the bacteria, you eliminate this source of trouble.

Employees like Elcide 75 because, when added to the emulsion, it is completely safe. In another recent test involving over 1000 machines, it was shown that Elcide 75 was safe, nontoxic, and harmless to sensitive skin. It also reduces the chance for bacterial infection caused by contaminated emulsion.

YOU KNOW WHAT YOU GET WHEN YOU BUY ELCIDE 75

With Elcide 75, you know you have the proper amount for effective treatment because you add it to your emulsions right in your plant. The standard recommendation is one ounce of Elcide 75 per 4 gallons of oil-water emulsion.

PROVE IT TO YOURSELF... TRY ELCIDE 75 IN YOUR PLANT

The best way to determine the value of Elcide 75 is to try it under your normal plant conditions, using your regular oil-water emulsion. After you've had an opportunity to compare results, we believe you'll agree that Elcide 75 is a valuable discovery that will greatly increase your plant efficiency.

PRODUCT SPECIFICATIONS ELCIDE 75

(Lilly's brand of bactericide for cutting fluids)

Active Ingredients—Thimerosal (Sodium Ethylmercuri Thiosalicylate) and Sodium o-phenylphenate.

Package	Price per Gal.
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5-gallon polyethylene . . .	$8.00
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Terms: 1%, 10th prox.; net, 30th prox. F.O.B. Indianapolis. Transportation paid via carrier of our selection.

For further information or to place your order, phone or write.

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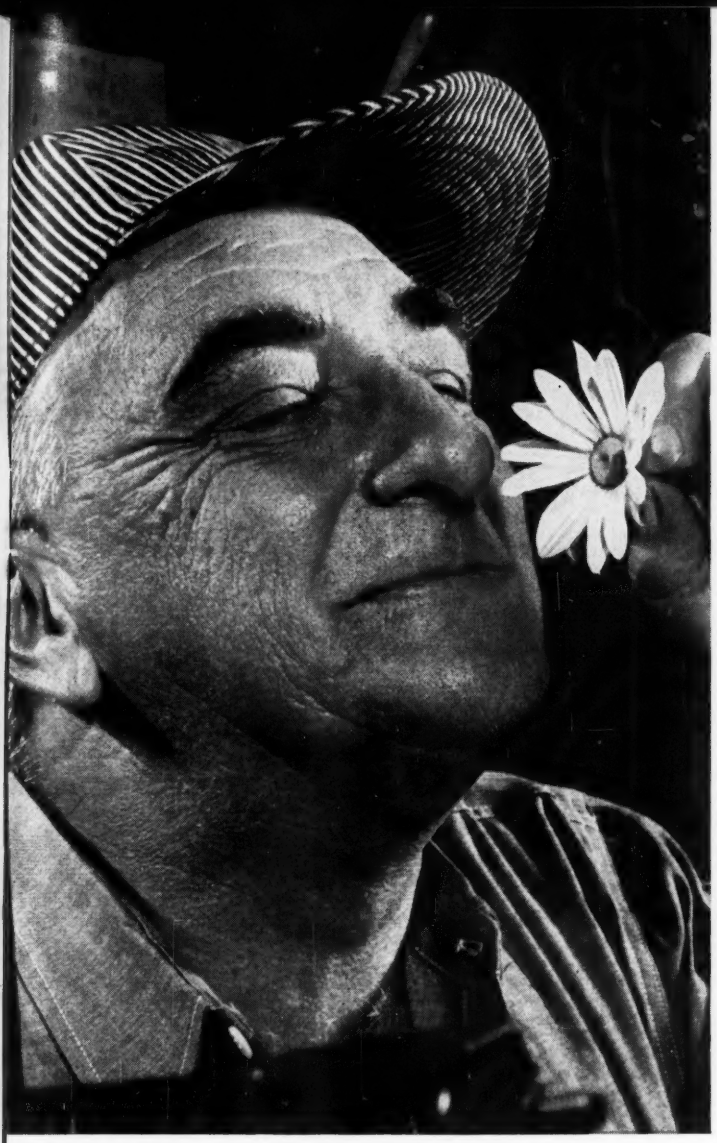
Lilly



DISCOVERED...A NEW WAY TO KEEP COOLANTS FRESH AS A DAISY!

Shop tests show ELCIDE 75 keeps cutting fluids fresh longer than ever before

Now . . . there's a new and better way to control the harmful bacteria that cause oil-water emulsion breakdown, acidic corrosion, and rancid odor. It's Elcide 75 . . . a product directly related to one of the most powerful bacteria killers used in medical science today . . . a material that has proved itself in the exacting field of surgical practice and has now been adapted to solve the problem of bacterial growth in soluble oil emulsions.

A large Midwestern manufacturing plant tested Elcide 75 under actual job conditions.

They found that one ounce of Elcide 75 added to each 4 gallons of oil-water emulsion . . .

extended useful life of the emulsion 5½ times . . .

greatly reduced cutting oil requirements . . .

delayed odor and emulsion break for 22 weeks, while untreated machines had to be dumped at the end of four weeks . . .

proportionately reduced down time for recharging . . .

was safe and nontoxic . . .

eliminated bacterial slime and reduced acidic corrosion . . .

drastically reduced their disposal problems.

WHAT ELCIDE 75 CAN DO FOR YOU

This combination of inhibitors can extend the life of your oil-water emulsions 2 to 4 times, or more, and can dramatically cut your oil-concentrate purchases. At the same time, production will increase because you have less down time for recharging.

ELI LILLY AND COMPANY · AGRICULTURAL AND INDUSTRIAL PRODUCTS DIVISION
INDIANAPOLIS 6, INDIANA TELEPHONE: MELROSE 6-2211

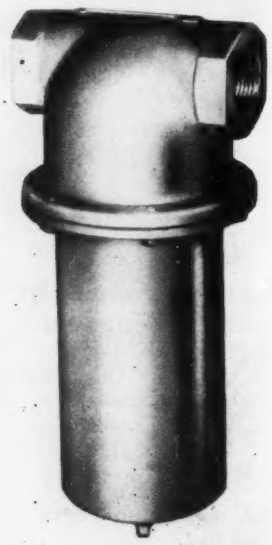
NEW PRODUCTS

FELT FOR REVERSE JET AIR FILTRATION MEDIA

Two special new felts, one made of Dacron, the other of Orlon, engineered especially for use in reverse jet air filtration media, are now being offered by American Felt Co., Glenville, Conn. Special features attributed by the manufacturer to the new items are: dimensional stability (less than 2½ percent shrinkage in either direction determined by commercial laundering tests), exceptional abrasion resistance, uniform air flow, good flex life and acid-alkali resistance. Samples of these felts and a data card giving full specifications of the fabrics may be obtained by writing for the folder describing Dacron Mdse. 61DA18 Special and Orlon Mdse. 610R18 Special directly to American's home office in Glenville or to any of its sales offices throughout the country.

HYDRAULIC FILTER

A new line type hydraulic filter has been developed by Capital Engineering & Mfg. Corp, for filtering of all types of



hydraulic fluids, lubricants, fire-resistant liquids and coolants. The filter is smaller in size than others of equal filtering capacity, reportedly making its use possible in restricted spaces where other cannot fit. The filtering element is made from Monel wire mesh in small corrugations with Monel support with top and bottom of Monel metal. This element is fitted into a housing of aluminum alloy and is recommended for pressure of 175 psi. It is quickly and easily disassembled for easy cleaning and reassembled in a minimum of time. For further information, write Capital Eng. & Mfg. Corp., 5837 S. Ashland Ave., Chicago 36, Ill.

TOOL KIT CONTAINS MAGNETIC CHUCK

An easy-to-use, economical tool kit containing a demountable precision diamond wheel and magnetic chuck plus all the accessories for quickly and easily finishing grinding carbide cutting tools to provide finer finishes, sharper cutting edges, increased tool life and faster cutting is now available from Penn Scientific Products Co., Abingdton, Pa. The magnetic chuck, which can be used with Pensco single layer, triple layer or impregnated diamond grinding rings for smooth-cushioned precision grinding, is adaptable to all 6 inch carbide tool grinders, holds diamond grinding rings securely and accurately in place and permits instant simple changing of grinding rings with no additional truing-up of the wheel. The low cost Pensco diamond grinding wheel rings are in various grit sizes, instantly snap into place ready for use and are said to be practically indestructable, long lasting and inexpensive to replace.

For full details, write Penn Scientific Products Co., Abington, Pa.

MODEL MIST COLLECTOR

A compact, efficient mist collector has been introduced as part of Torit Manufacturing Company's new line of mist and fog collecting equipment. The new mist collector, MC-412, is especially designed to control mists from wet machining operations that can endanger employee health or create fire and safety hazards. Measuring only 22 in. square, it can be floor-mounted on legs or mounted on walls or ceiling to conserve space. It measures 47¼ in. high, including its top-mounted motor and blower. The new collector requires no high voltage electronic equipment, the company says, but filters mist and fog by passing the air through high-performance spun glass filters that can be easily and economically replaced. In normal operation, the filters will operate efficiently for many months.

In addition, condensed coolant mist is collected in the bottom of mist collector where it can be drained for re-use. By means of a connecting tube, the MC-412 system can be connected directly to the machine coolant system for continuous flow. Additional information on the MC-412 mist collector is available from Torit Manufacturing Company, St. Paul 2, Minn.

Lubriplate No. 630-2 is a high temperature, extreme pressure, water-repellent, grease type lubricant. Ideal for the general lubrication of Industrial, Automotive, Construction, Farm and Marine Equipment. Lubriplate Grease Gun Cartridges provide an easy, quick, economical means of application. Prevent the waste and mess of hand filling. Packed 10 Cartridges in a handy carrying carton.

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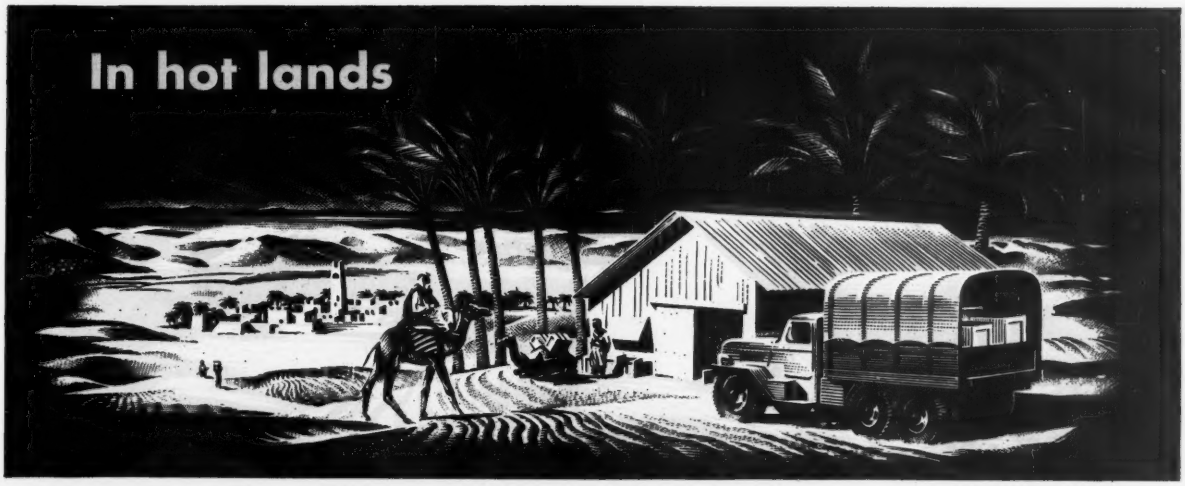
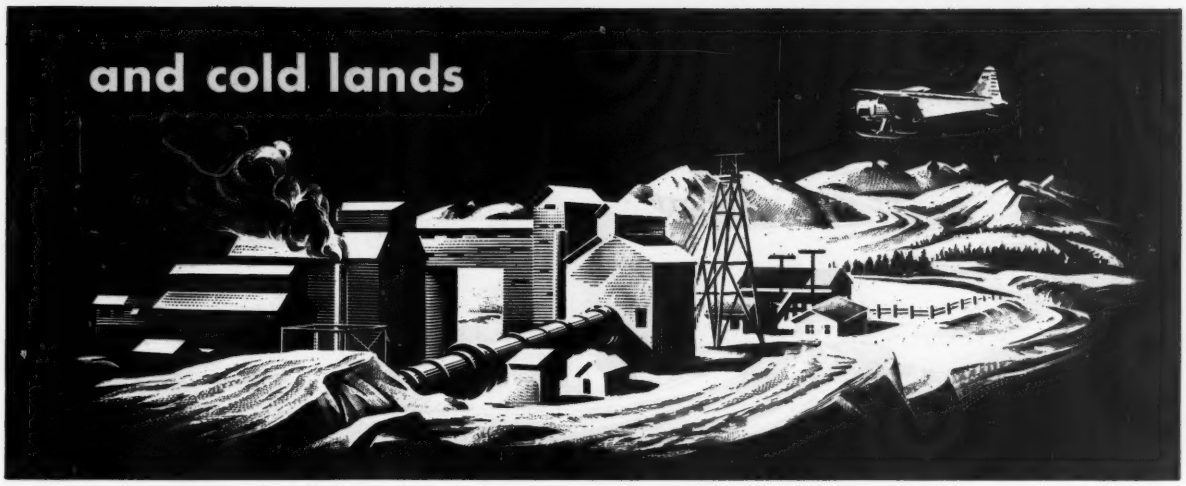
For nearest LUBRIPLATE distributor see Classified Telephone Directory. Write for free "LUBRIPLATE DATA BOOK"... a valuable treatise on lubrication. LUBRIPLATE DIVISION, Fiske Brothers Refining Company, Newark 5, N. J. or Toledo 5, Ohio.





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Shell Tellus Oils—for closed hydraulic systems

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Shell Macoma Oils—for extreme pressure industrial gear lubrication

Shell Voluta Oils—for high-speed quenching with maximum stability

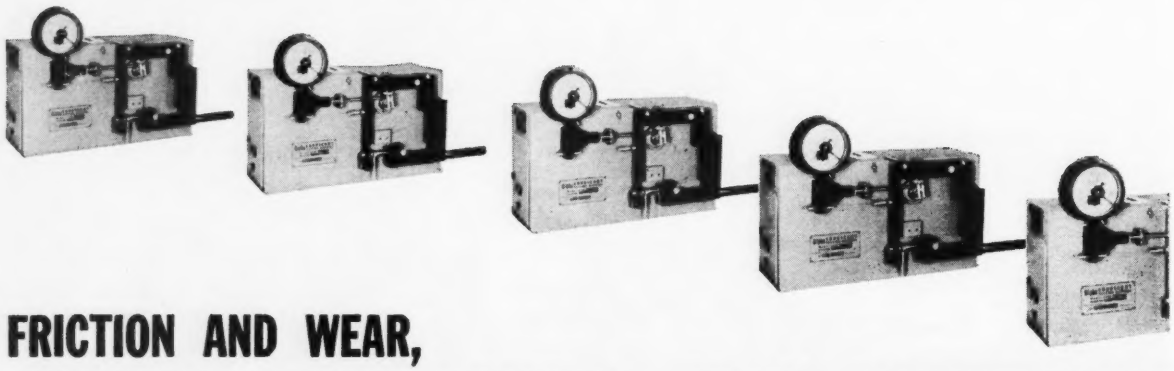
Here's a grease you can count on to remain plastic in sub-zero weather and, at the same time, remain stable under sustained high temperatures. On job after job, Shell Alvania Grease has successfully replaced dozens of special lubricants.

Alvania® Grease also has an outstanding performance record on the toughest anti-friction bearing grease applications. It is ideal for wet, humid applications because it is inhibited to prevent water corrosion. It gives good lubrication under conditions which normally spell trouble.

The world-wide availability of Alvania Grease is assurance that your customers abroad will get the same performance from your equipment that domestic customers rely upon. For complete information on this truly multi-purpose grease, write Shell Oil Company, 50 West 50th Street, New York 20, N. Y., or 100 Bush Street, San Francisco 6, California.

SHELL ALVANIA GREASE

SHELL



FRICTION AND WEAR, LIKE METAL FATIGUE PROBLEMS, DEMAND COMPREHENSIVE TESTING TO BREAK DOWN THE APPARENT COMPLEXITY

Complete isolation of individual effects is necessary to determine the influence of each condition on boundary lubrication

Although many complex phases of engineering have something in common, there is an uncanny parallel in the study of frictional effects and fatigue effects in metals.

The complexity of fatigue of metal problems involved such effects as metallurgical composition, geometrical shape, size, surface geometry, residual stresses, atmospheric environment, metal working and assembly-induced stresses. Since little was known about the full influence of any of these conditions on fatigue, full-scale simulated service testing of structures and assemblies allowed only composite analyses of these effects. In these composite analyses, however, large variations existed and researchers were unable to pinpoint the particular effect which accounted for the variations.

TESTING IN DETAIL PRODUCED RESULTS

With the advent of World War II, aircraft and military developments forced a more systematic approach to the problem of metal fatigue. A large number of small and relatively inexpensive testing machines were installed to investigate individual effects which were known or presumed to make up the complexity. This approach produced results.

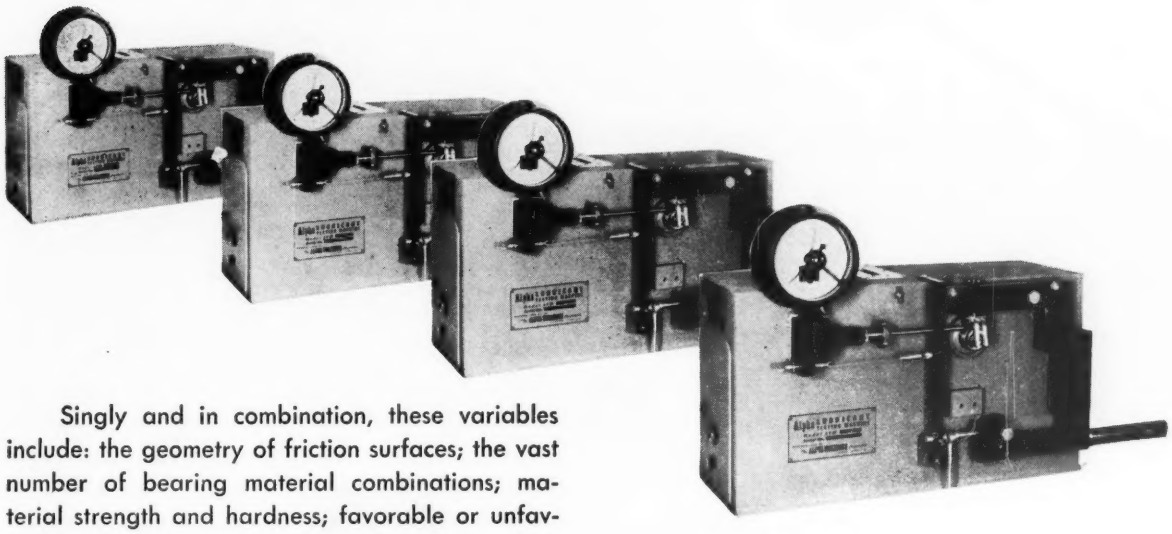
Concentrating on metallurgical composition brought about a knowledge of the decrementary effect of inclusions near the surface. Tests concentrating on geometrical shape disclosed reductions of fatigue strength due to abrupt changes of specimen contour. Surface geometry tests showed the strength-reducing effects of sharp notches. Tests for residual stresses revealed the beneficial effects of surface compression stresses and the detrimental effects of surface tension stresses. Atmospheric environment tests showed the strength-reducing effects of even minor surface corrosion.

Full-scale testing could never have isolated these effects. It wasn't until millions of hours of testing were conducted on small machines that many of the complex phases of fatigue life were cleared up. This made full-scale testing productive and economical.

SAME APPROACH NEEDED FOR FRICTION AND WEAR EFFECTS

The complexity of boundary lubrication rivals the complexity of metal fatigue. It needs, and will receive, increasing attention! Millions of hours of testing must go into the effort to isolate and determine the significance of a multitude of damaging or beneficial effects.

THE BETTER THE TESTING EQUIPMENT . . .
THE LOWER THE COST OF PRODUCTIVE RESEARCH



Singly and in combination, these variables include: the geometry of friction surfaces; the vast number of bearing material combinations; material strength and hardness; favorable or unfavorable formation of surface deposits; surface oxidation; the influence of frictional heat on the bearing metals and the lubricant; bearing pressures; sliding velocity; and so on.

BASIC PROGRAM ALREADY BEGUN

Because of commercial unavailability, we have found it necessary to develop our own research tools for studies in detail. For low velocity EP work, our LFW-1 testing machine, illustrated above, is exceptionally accurate and flexible. We have 16 of these machines operating on a 24-hour-a-day basis, and 42,000 testing hours to date on these machines have resulted in the development of several new lubricant compositions with exceptional characteristics. LFW-1 machines can now be purchased from The Alpha-Molykote Corporation.

SEND FOR FREE LITERATURE

Our Bulletin #106 describes in detail the Alpha Model LFW-1 Lubrication-Friction-Wear Testing Machine. If you have not already received a copy, we will send one to you by return mail.

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INDUSTRY'S MOST VERSATILE LUBRICANT
MOLYKOTE

The **ALPHA-MOLYKOTE** *Corporation*

Main Factories: 65 Harvard Ave., Stamford, Conn.
71 Arnulfstrasse, Munich 19, Germany

PERSONALS

The appointment of John A. Dersch as Director of Engineering for the Read Standard Division of Capitol Products Corp., York, Pa., has been announced.

L. M. Miller has been promoted to the newly created position of sales manager for the aliphatic organic derivatives department of the Armour and Company Chemical Division.

Peter H. Lauer has been elected treasurer of Flexonics Corporation. Lauer succeeds R. R. Muller, new general manager of the company's aeronautical Division.

C. Perry Streithof has been appointed director of the central engineering department of American Viscose Corporation.

Edwin M. Strom, for the past two years assistant manager of the graphite and lubricants division of the Joseph Dixon Crucible Company, Jersey City, N. J., has been promoted to manager of the division. Strom will continue to give primary attention to the further development and field testing of Dixon natural colloidal graphite, a new product developed for the first time in the United States by Dixon. Also at Dixon, Ralph Rathyen has been promoted to the post of industrial sales manager. In his new position, Rathyen will head up the sales activities of the paints, crucible and refractories, and graphite and lubricants divisions.

William K. Lambert has been named comptroller of Parker Aircraft Company's four Los Angeles divisions engaged in the design, development, testing and production of fluid handling parts, components and subsystems.

Three new salesmen have been appointed to the sales staff of Jefferson Chemical Co., Inc. Robert E. Lee and Patrick W. Loftus will report to the company's Chicago office and Ellsworth F. Benson will report to the New York office.

INDUSTRY NOTES

Manufacturing and administrative facilities of the Mettler Company, Inc., division of Eclipse Fuel Eng. Co., have been moved from Los Angeles to Rockford, Ill. The move was made in order to bring Mettler, major manufacturer of packaged, fan-type power and process burners, closer to its major markets.

Pennsalt Chemicals Corporation has announced the establishment of a new sales office for its corrosion engineering products in the Kansas City, Mo., area under the management of Edward H. Garnett of the Shutt Process Equipment Corporation.

Natural Colloidal Graphite

PERFECTED!

99.75% Minimum Purity! (1)

95% Minimum Stability! (2)

Dixon Suspension "10", natural colloidal graphite in oil, possesses such extreme fineness and stability that it cannot be removed by even 9 thicknesses of Whatman No. 42 Filter Paper. Wool felt filters and automotive filters also will not remove Suspension "10".

Dixon Suspension "10" has a wide range of high temperature and extreme pressure applications, including . . .

- *glass molds*
- *high temperature chains*
- *forging*
- *die-casting*
- *extruding*
- *kiln car bearings*

It has been approved for use in automatic lubricating systems.

(1) Firing at 1500°F.
(2) Eleven minute centrifuging at 0.02% dilution.

■ Technical literature and engineering assistance available.

DIXON

JOSEPH DIXON CRUCIBLE COMPANY, JERSEY CITY 3, N. J.
Graphite and Lubricants Division,

meet high performance specifications

- Withstand higher operating temperatures (Up to 75° hotter than lithium based greases).
- Are water resistant (GA-10 greases are insoluble in water).
- Are resistant to radiation damage from X-Rays, gamma rays, beta rays and neutrons (GA-10 greases maintain good gell structure up to 500 megaroentgens of gamma rays). Probably one of the most radiation resistant products yet discovered.
- Give improved bearing performance (last as much as 2-3 times longer than ordinary soap based greases).
- Have superior work stability (hold up better under severe operating conditions).
- GA-10 greases have higher ASTM dropping points (in excess of 580°F).
- Are compatible with other types of greases.
- Have excellent pumpability in either pressure or automatic feed systems (GA-10 greases are faster flowing than most soap gelled greases).

The unique properties of Oronite GA-10 gelling agent make possible the production of superior high-performance grease lubricants for a variety of applications. Whether you make multipurpose grease lubricants, automotive, aircraft, marine, nuclear reactor, or special purpose greases you can now make them better with Oronite GA-10.

Call or write the Oronite office nearest you for detailed information.
Send for technical bulletin describing properties of Oronite GA-10.

ORONITE CHEMICAL COMPANY
A CALIFORNIA CHEMICAL COMPANY SUBSIDIARY
EXECUTIVE OFFICES • 200 Bush Street, San Francisco 20, California
SALES OFFICES • New York, Boston, Wilmington, Chicago, Cincinnati, Cleveland, Houston, Tulsa, Los Angeles, San Francisco, Seattle
Foreign Affiliate: California Chemical International, Inc., San Francisco, Geneva, Panama

4874

NEW PRODUCTS

HI-TEMP HYDRAULIC FLUID AND LUBRICANT BASE

Recently announced by Dow Corning Corporation, Midland, Michigan, is a new synthetic fluid, an alkyl silane, which holds promise as a hydraulic fluid for use in systems subjected to temperature and environmental extremes including those associated with high performance aircraft and missiles. Designated as QF-6-7009, this new material is said to have better lubricating properties than most previously available high temperature fluids. It is thermally stable in closed systems over the wide range of —25 F to 550 F for long periods; up to 700 F for short times. In addition to its use as a hydraulic fluid, the fluid has shown promise as a base oil in high temperature turbine lubricants.

SYNTHETIC CHAIN LUBRICANT

The Industrial Lubricant Division of Lehigh Chemical Company of Chestertown, Maryland is now marketing a new synthetic chain lubricant, Anderol L-732. This new material reportedly is effective in overcoming the many problems previously associated with chain lubrication. Unlike the light penetrating oils commonly used to lubricate chains, L-732 is a tacky, semi-fluid grease which will flow to some extent, but retains a gel structure that lubricates as a grease, giving much longer lubrication periods. Also, being tacky, it completely resists wipe-off, throw-off, and run-off . . . and it will remain effective from six months to over a year.

Lehigh chemists have built many unique properties into this synthetic lubricant. Being a synthetic, L-732 will not gum nor will it leave a varnish residue. In fact, L-732 will dissolve most deposits which have already formed on the chain. The lubricating grease penetrates quicker than a petroleum oil, insuring complete lubrication throughout the chain.

In application, the L-732 need only be applied in a thin stream on only one side or top of the entire length of chain. When the chain is run, the lubricant will work all through the chain, leaving a thin film over all parts, both inside and out. This film is an adequate rust preventive and will give effective protection for many months because of its resistance to sling-off, its good lubricity and nonvolatile characteristics, its oxidation stability, and the fact that it is not affected by moisture.

For more information on this new long-life chain lubricant write Lehigh Chemical Company, Industrial Lubricant Division, Chestertown, Maryland.

NEW LINE OF FREON HOSE AND REUSABLE COUPLINGS

A valuable contribution to simplified installation and servicing of automotive air conditioning and refrigeration systems is provided by the new line of Freon 12 hose and reusable couplings being introduced by The Imperial Brass Mfg. Company of Chicago. To eliminate costly inventories of made up hose assemblies, Imperial is offering the hose in 50 feet coils together with a variety of hose couplings, so that any desired hose assembly can readily be made up. Hose couplings are designed so that they can be used over and over. Imperial's Freon 12 hose is designed to operate satisfactorily in the temperature range of —20° F to +250° F, with recommended maximum working pressure of 350 psi. Hose is available in five sizes from 3/16 in. to 5/8 in. ID. The hose has an inner seamless tube of synthetic rubber, reinforced with cotton braid, plus an outer cover of cotton braid impregnated with synthetic compound. Imperial's cadmium plated steel reusable couplings are designed to assure maximum flow. Pipe thread ends are the Dryseal type. Female 45° flare swivel, male 45° flare connectors, and male pipe thread connectors are available in the standard sizes. A complete package assortment of couplings and hose and assembly mandrels can be obtained together with a 21 drawer steel cabinet that fills all the needs for fast, correct service on automotive and bus air conditioning systems, refrigerated trucks, pullout units, overnight charging lines for dairy trucks, truck refrigeration plates and vibration absorption lines. Stock unit, including cabinet, is designated No. AC-100 and contains 70 reusable couplings, 50 feet each of three sizes of hose and two assembly mandrels. For more information write The Imperial Brass Mfg. Co., 6300 W. Howard St., Chicago 48, Ill. and request Bulletin No. 3084.

POLYETHYLENE FUNNEL

A linear polyethylene Buchner funnel, designed for the filtration of corrosive slurries, has been announced by the American Agile Corp. Made of high density polyethylene, the funnel is equipped with an inclined bottom to provide complete drainage; it is also fitted with a polyethylene drain spigot. This new product is 20 in. in diameter, stands 22 in. high and is made from ¼ in. thick material. The filter plate is ½ in. thick and is supported. The plate has been drilled with 1/16 in. holes on ⅜ in. centers. For further information, write the American Agile Corp., P. O. Box 168, Bedford, Ohio.

YOUR LUBRICATION PROBLEM IS OUR BUSINESS

Call in one of our specialists!

Contact the nearest member or representative of the Far-Best Group when you need:

Cutting Oils
Tapping Compounds
Drawing Compounds
Rolling Oils
Metal Cleaners
Sulfonated Oils
Heat Treating Compounds
Grinding Coolants
Rust Preventatives
Lubricant Greases
Lubricant Additives
Motor Oils
Specialized Lubricants
Lanolin
Industrial Soaps
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The Far-Best Corporate Group includes:

- **Allube Corp., 928 Allen Avenue Glendale, Calif. Chapman 5-1158**
  —33 years manufacturing experience
- **Far-Best Corp., 6715 McKinley Ave., Los Angeles, Calif. Pleasant 8-3181 and 2800 Commerce St., Franklin Pk., Ill.**
  —26 years manufacturing experience
- **O. L. King & Co., 782 Minnesota Street, San Francisco, Calif. Valencia 6-6701**
  —26 years manufacturing experience
- **Dacus Oil Corp., 782 Minnesota St., San Francisco, Calif. Valencia 4-5185**
  —25 years manufacturing experience
- **Sanford Process Corp., 6920 S. Central Ave., Los Angeles, Calif. Ludlow 3-4856**
  —10 years development and production experience

A total of 120 years' industrial experience in solving production and maintenance problems.

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Pacific Machinery & Tool Steel,
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Seattle, Washington

Far-Best, Inc.,
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AREA REPRESENTATIVES

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Chicago/Milwaukee	J. N. Waddell, Henry H. Cross Co., 36 So. Wabash Ave., Chicago, Ill.
Cleveland/Youngstown	A. H. Turner, Standard Oil Co., Midland Bldg., Cleveland 15, Ohio
Columbus	H. W. Irion, National Refining Co., 1885 Coventry Rd., Columbus 12, Ohio
Dayton/Cincinnati	G. H. Leffel, D. A. Stuart Oil Co., 1021 N. Limestone St., Springfield, Ohio
Houston/Dallas/Fort Worth	R. N. Golden, Sun Oil Co., P. O. Box 2880, Dallas 21, Texas
Hudson Mohawk/Rochester/Buffalo	D. F. Wilcock, General Electric Co., 1 River Rd., Schenectady, N. Y.
Indianapolis/Fort Wayne	F. Pusey, 828 No. Graham Ave., Indianapolis 19, Ind.
Intermountain	D. P. Thomas, Columbia Geneva Steel Div., P. O. Box 510, Provo, Utah
Kansas City/St. Louis	G. W. Gergeceff, Granite City Steel Co., Madison Av. & 20th St., Granite City, Ill.
Los Angeles	R. C. Dishington, General Petroleum Corp., 2401 E. 27th St., Los Angeles 54, Cal.
Pittsburgh/Wheeling	L. B. Sargent, Alcoa, P. O. Box 772, New Kensington, Pa.
Puget Sound	J. H. Hood, Boeing Airplane Co., Seattle, Wash.
Twin Cities/Duluth/Iron Range	R. O. Erickson, Shell Oil Co., 5516 Glegarry Pkwy., Minneapolis 24, Minn.

SECTION	CHAIRMAN	SECRETARY	MEETING—DATE & PLACE
Baltimore	C. J. Devries	H. D. Plumly, National Refining Co., 1325 E. 35th St., Baltimore, Md.	3rd Thursday—Hotel Stafford, Charles & Madison Sts., Baltimore, Md.
Boston	W. F. Sentner	E. E. M. Schreiter, 179 Highland Ave., Arlington 74, Mass. MI 3-6551 (Arlington, Mass.)	3rd Monday—M.I.T. Faculty Club, Memorial Drive, Cambridge, Mass.
Buffalo	R. C. Sager	D. B. Capwell, Esso Standard Oil Co., P. O. Box 95, Kenmore 17, N. Y.	3rd Wednesday—Hotel Sheraton, Delaware Ave., Buffalo, N. Y.
Central Ohio	C. M. Allen	R. H. Roby, Timken Roller Bearing Co., 1025 Cleveland Ave., Columbus, Ohio	4th Monday—Batelle Memorial Institute, Columbus, Ohio
Chicago	E. W. Drummond	J. E. Sonnefield, Alcoa, Madison & Mannheim Road, Ill. ES 8-7171	3rd Thursday—228 North LaSalle St., Chicago, Illinois
Cincinnati	E. L. Curtis	D. H. Popma, J. E. Seagram & Sons, Inc., Lawrenceburg Ind. Lawrenceburg 700	2nd Monday—Engineering Society Hdqtrs., East McMillan St., Cincinnati, Ohio
Cleveland	G. L. Smith	L. O. Ray, Alcoa, 2210 Harvard Ave., Cleveland 5, Ohio	4th Tuesday—Cleveland Engineering Bldg., 3100 Chester Ave., Cleveland, Ohio
Connecticut	L. Edwards	J. Kerruish, Jr., Esso Standard Oil Co., 66 Thompson St., Hamden 14, Conn. JA 7-4183 (Hartford, Conn.)	3rd Wednesday—Sun Valley Acres, Rte. 6A, Meridan, Conn.
Dallas—Fort Worth	E. B. Palmer	R. J. Neely, 3025 Bigham St., Fort Worth 16, Texas PE 7-0760	4th Thursday
Dayton	H. L. Sellers	C. R. Chaplin, Standard Oil Co. (Ohio), 1135 Warder St., Springfield, Ohio, FA 4-3575	4th Tuesday—Engineers Club of Dayton, 112 Monument Ave., Dayton, Ohio
Detroit	C. F. Cowley	W. J. Wejtowicz, H. A. Montgomery Co., 17191 Swift Ave., Detroit 3, Mich. JO 4-6412	Staggered—Engineering Society of Detroit, East Farnsworth at John Road, Detroit, Mich.
Duluth—Iron Range	C. D. Johnson	W. Erickson, 19 Battlefield, Stoney Creek, County of Wentworth, Ontario, Canada	
Fort Wayne	C. R. Griffith	J. M. Olson, 6606 Fernwood Drive, Fort Wayne, Ind.	
Houston	T. L. Gordy	E. Shannon, Humble Oil & Refining Co., P. O. Box 2180, Houston, Texas. JA 9-4141 Ext. 378	1st Monday
Hudson—Mohawk	J. P. Thomas	R. A. Reissig, 23 Glendale Ave., Delmar, N. Y. 4-2111 (Albany, N. Y.)	3rd Wednesday—Crossroads Restaurant, Latham, N. Y.
Indianapolis		W. J. Adkins, 1731 No. Colorado, Indianapolis, Ind.	
Intermountain	J. E. Bradley, Jr.	J. A. Kyle, Sinclair Oil Refining Co., P. O. Box 248, South Salt Lake City, Utah	2nd Monday—Aviation Club, 6th East & South Temple, Salt Lake City, Utah
Kansas City	C. L. Austin	J. L. Taylor, Socony Mobil Oil Co., 925 Grand Ave., Kansas City, Kan.	
Kingsport	H. E. Goddard	P. J. Galbreath, Tennessee Eastman Co., Kingsport, Tenn. CI 5-6373	Staggered Fridays
Los Angeles	R. Wheeler	L. P. Richardson, Richfield Oil Co., 932 Janet Lane, Lafayette, Cal. MA 9-4111	3rd Wednesday
Louisville	J. K. Wyatt	R. E. Butler, 4811 Kingfisher Way, Louisville, Ky.	
Milwaukee	R. Rheingans	F. Johnson, Standard Oil Co., 2533 No. 111th St., Milwaukee 10, Wisc. GL 3-3509	3rd Thursday—Engineers Society of Milwaukee, 3112 W. Highland Blvd., Milwaukee, Wisc.
Montreal	J. Ussher	J. H. Woodward, Woodward, Lemieux & Saunders, Ltd., 4616 St. Catherine St. W., Montreal, Que., Can. GL 4814	1st Wednesday—LaSalle Canadienne, Canadian Pacific Railway, Windsor Station, Montreal, Que., Can.
New York	H. D. Lewis	S. E. Manilych, American Oil & Supply Co., 238 Wilson Ave., Newark 5, N. J.	Staggered—Schrafft's, 556 Fifth Ave., New York City, N. Y.
Northern California	G. H. Hommer	J. L. Dreher, California Research Corp., P. O. Box 1627, Richmond, Calif. BE 2-1514, Ext. 2448	2nd Thursday
Ontario	W. H. Mann	A. G. Almack, Imperial Oil Ltd., 150 Laird Dr., Toronto 17, Ont., Can. HU 9-2121	2nd Tuesday
Philadelphia	N. A. Bracht	L. E. Preston, Cities Service Oil Co., 117 So. 17th St., Philadelphia 3, Pa.	3rd Thursday—D. F. McAllister & Sons, 1811-17 Spring Garden St., Philadelphia, Pa.
Pittsburgh	A. J. DeArdo	R. C. Garretson, Westinghouse Electric Corp., 2515 Collins Rd., Pittsburgh 35, Pa.	2nd Thursday—Royal York Hotel, Pittsburgh, Pa.
Puget Sound	C. S. Armstrong	F. Stewart, 515 No. 104th St., Seattle 33, Wash.	2nd Tuesday—711 W. Elliott Ave., Seattle, Wash.
Richmond	O. L. Miller	J. E. Connelly, 102 Oakwood Ave., Hopewell, Va.	2nd Monday
Rochester	C. L. Miller	C. E. Snodderly, 138 Hyde Parkway, Palmyra, N. Y.	2nd Wednesday—German Club, 315 Gregory St., Rochester, N. Y.
Saginaw Valley	J. M. Small	D. H. Smallridge, Dow Chemical Co., 9-314 Building, Midland, Mich.	4th Thursday—Zender's, Frankenmuth, Mich.
St. Louis	E. R. Harris	J. H. Koester, 9730 Colony Drive, St. Louis 15, Mo. GL 2-1100, Sta. 327	3rd Tuesday—Coronado Hotel, St. Louis, Mo.
Syracuse	H. E. Kaye	M. Michlin, 23 E. Elizabeth St., Skaneateles, N. Y.	3rd Monday—Yates Hotel, Syracuse, N. Y.
Twin Cities	D. J. Armstrong	R. J. Eischens, Northern States Power Co., 1600 Chestnut, Minneapolis, Minn. FE 8-7292	Last Thursday—Dyckman Hotel, Minneapolis, Minn.
Wheeling	S. R. Fredlock	W. L. Cowan, American Oil Co., Wheeling, W. Va.	
Youngstown	E. M. Downs	W. A. Stranko, Market Street Extension, No. Lima, Ohio KI 9-3740 (Youngstown, Ohio)	2nd Thursday—Mahoning Valley Country Club, Girard, Ohio

One of the 3 essentials for grease manufacturers is EXCELLENCE . . .

Number 4 of a series

BARAGEL* provides EXCELLENCE

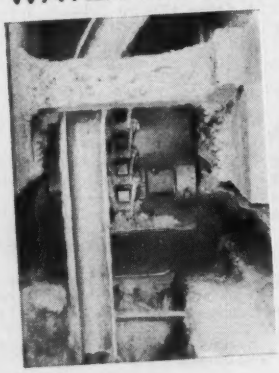
BARAGEL provides EXCELLENCE because of . . .

1. LONG LIFE . . . greases compounded with BARAGEL . . . last longer . . . require less frequent application.
2. WATER RESISTANCE . . . greases compounded with BARAGEL . . . do not wash out when submerged in water.
3. HEAT STABILITY . . . greases compounded with BARAGEL . . . do not melt even when exposed to extreme heat.

LONG LIFE

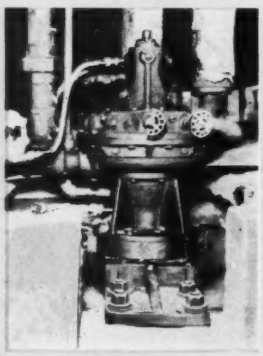
Here are conveyor troughing idlers on a feed belt assembly. By using BARAGEL grease . . . less lubricant is required . . . lubrication was reduced from daily basis to once every 3 months.

WATER RESISTANCE



Bearings on the shaft of this drag box are completely submerged in water at all times. Use of BARAGEL grease, under such conditions, proved far superior to any other type previously used.

HEAT STABILITY



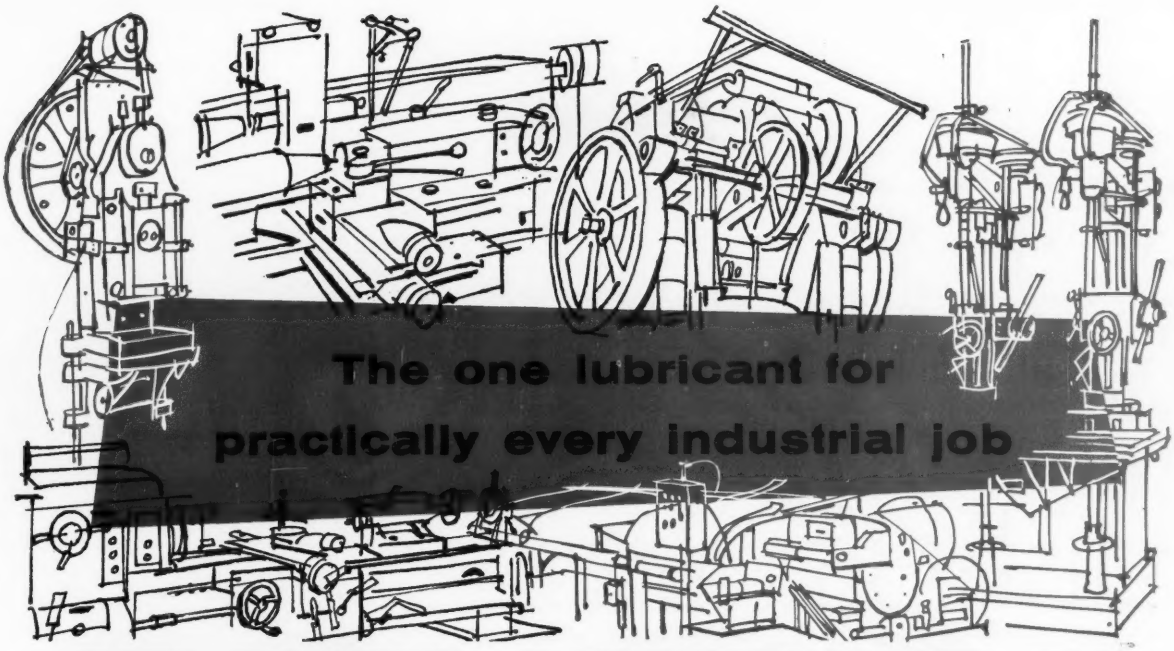
A high temperature application of BARAGEL grease used on all bearings in steam drive turbines. Lubrication was reduced from daily to weekly basis. BARAGEL grease did not run at 320°F turbine temperature . . . at speeds of 4,000 to 5,000 rpm.

BAROID CHEMICALS, INC.

A SUBSIDIARY OF NATIONAL LEAD COMPANY

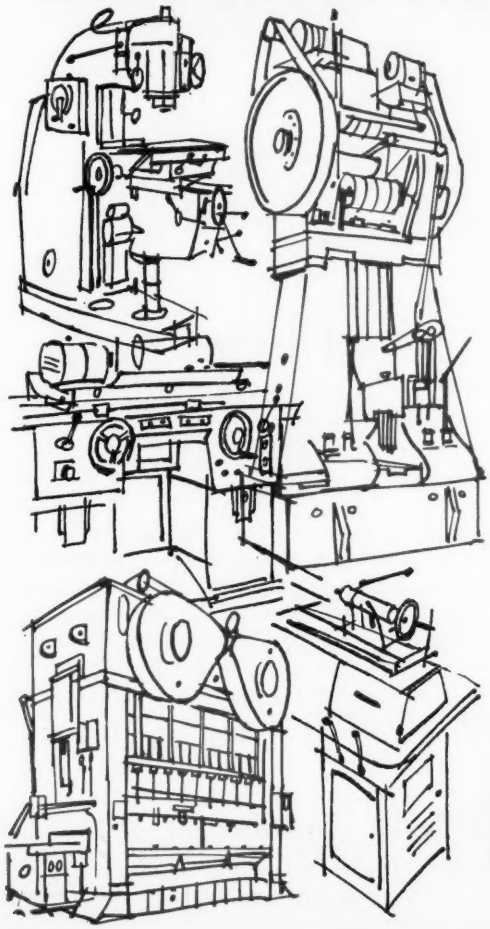
1809 SOUTH COAST LIFE BLDG. HOUSTON 2, TEXAS

*Trademark of National Lead Co.



ATLANTIC LUBRICANT 54

Performs on all types of grease-lubricated machinery • Cuts cost • Reduces lubricating errors



Atlantic Lubricant 54—an outstanding multi-purpose lubricant—assures:

- **Wider application**—almost universal application to grease-lubricated equipment in plants of all sizes
- **Excellent performance** at very high to extremely low operating temperatures
- **Reduced losses** from lubricating error
- **Lower costs** of warehousing and handling
- **Outstanding rust protection** for all ferrous surfaces
- **Unusual stability** against oxidation and softening—even under very rugged conditions
- **Easy, positive identification**—thanks to Atlantic Lubricant 54's distinctive purple color

Available from conveniently located warehouses. Call any of the Atlantic offices listed below.

ATLANTIC

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PROCESS PRODUCTS

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SYRACUSE, N.Y.
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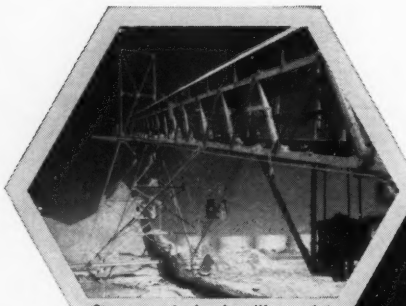
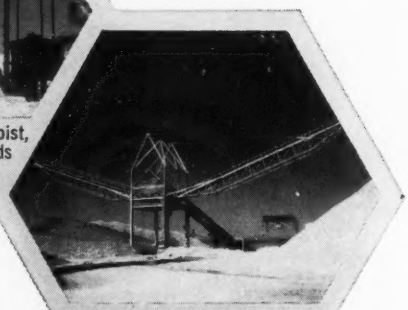
CHARLOTTE, N.C.
1112 South Boulevard

In South America
Atlantic Refining Company of Brazil, Rio de Janeiro

"In 30 months...not a single bearing lubrication failure with lithium-base grease!"



Conveyor-stacker handling moist, sticky material which builds up on the rollers.

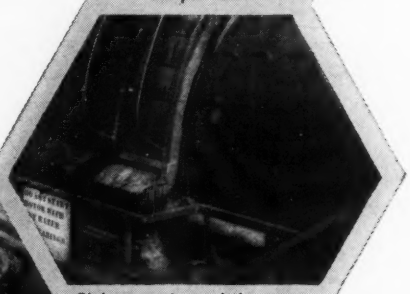


Ore unloading conveyor rollers handling 200 tons of ore per hour.

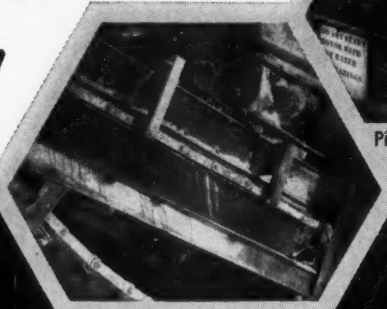
THE PROOF IS IN THE PERFORMANCE...

lithium-base grease does the job...

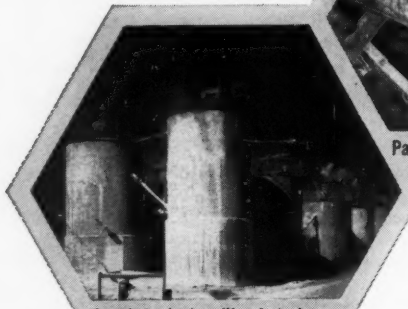
and does it better!



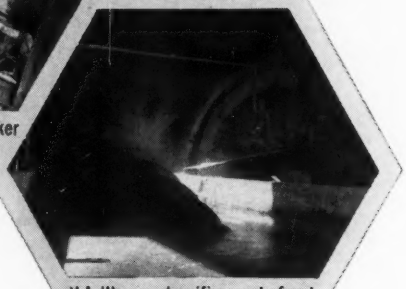
Pinion gear transmitting power from 600 h.p. motor to a ball mill.



Pan Conveyor handling hot clinker (1600°F), roller bearings in dusty, moist atmosphere.



Leach tanks handling hot slurry, agitators driven by Falk gear reducing units.



V.A. Wemco classifier, part of wet grinding system, lower bearing submerged in hot sludge.

Here's a report of our own experience with lithium-base grease under extreme industrial service conditions. Approximately 95% of the grease used in the plant of AMERICAN LITHIUM CHEMICALS, Inc., our subsidiary at San Antonio, Texas, is lithium-base, one-type grease. In thirty months operation we have not been able to trace a single cause of bearing failure to the lubricant used. The on-the-spot photos above give graphic evidence of the rugged bearing service requirements in this plant where lithium ores are processed into high-grade lithium hydroxide, itself an important ingredient in lithium-base grease. Performance like this is why grease chemists, manufacturers, marketers and users all attest to the superiority of lithium-base... the *one* grease in place of *many* for efficient and economical operation.

TRONA
REG. U.S. PAT. OFF.

Want to know more about TRONA lithium hydroxide monohydrate? Send for our technical bulletin on this important chemical ingredient in lithium-base greases.

American Potash & Chemical Corporation

3000 West Sixth Street • Los Angeles 54, California

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Lube Lines

by A. F. Brewer*

Deposits

To function effectively, a lubricant must be able to penetrate to and completely cover the moving surfaces which require lubrication. Any obstruction which can impede lubricant's flow can adversely affect the maintenance of the lubricating film.

Modern machinery is designed so that there will be no intentional obstructions in the lubricating systems. In fact, one of the essential functions of the break-in procedure is to wear down minute rough spots (asperities) on bearing, shaft and gear tooth surfaces so that fluid film lubrication can be maintained more dependably.

However, unintentional, though oftentimes unavoidable obstructions can develop during operation depending on the type of machine, severity of service, and temperature. Engines and compressors can be very serious offenders. Centralized pressure greasing systems adjacent to steel rolling mill machinery also can be a source of trouble. *Basically these obstructions result largely from deposits formed from, or as a result of the process of lubrication.*

HOW THEY AFFECT LUBRICATION

In a reciprocating engine or compressor where the compression or pressure is maintained by the flexibility, seating and fit of the piston rings, any accumulation of carbonaceous deposits in ring grooves could cause the rings to function sluggishly or to become stuck. Carbonaceous deposits result from the use of chemically unsuitable oils which are unable to withstand prevailing temperatures.

Carbonaceous deposits also can affect compressor output when they build up on discharge valves. Here they can become cumulative because the very high discharge temperatures which they induce can in turn lead to further carbon deposition from oil break-down.

Viscosity, carbon residue content and volatility are indicative characteristics which relate to carbon deposits. Normally oils of greatest volatility produce the least amount of carbon. As volatility relates to rate of evaporation and viscosity, lower viscosity oils are preferable because, as a rule, they have lower flash points and evaporate more cleanly. As the fire hazard under conditions of heavy deposit formation in compressors can be serious, considerable interest prevails in the use of wholly synthetic compressor lubricants. The very high autogenous ignition temperatures of these products indicate reduced flammability and renders them highly fire resistant.

Deposits can interfere with anti-friction bearing lubrication where limited or measured amounts of oil or grease are depended upon to maintain lubrication. In this case, penetration of dust particles past the seals through windage is the more prevalent offender, rather than temperature. Obviously any marked accumulation of such foreign matter can cause the rolling elements to function sluggishly. This can be the forerunner of ball or roller sliding which can soon call for bearing replacement.

HOW THEY DEVELOP

Temperature and entry of foreign solid matter have been indicated as prominent causes of deposits in lubricating systems. There can be other inherent contributing causes. For example:

1—Rust particles and paint scale.
2—Pipe scale, core sand and metal particles remaining from time of assembly.
3—Inadequate oil reconditioning.
4—Chemical effect of gas contamination.
5—Oxidizing effects of water and air at elevated temperatures.
6—Use of lubricants which are physically and chemically unable to withstand operating conditions.

Rust is the result of surface oxidation of iron or steel in the presence of moisture. It can develop from actual contact with water or from moisture in the air.

To correct for or reduce the tendency towards rusting, surfaces should be kept as dry as possible and a rust inhibiting lubricant which—by reason of its polarity—will provide a protective oil film capable of resisting penetration or washing by water should be used.

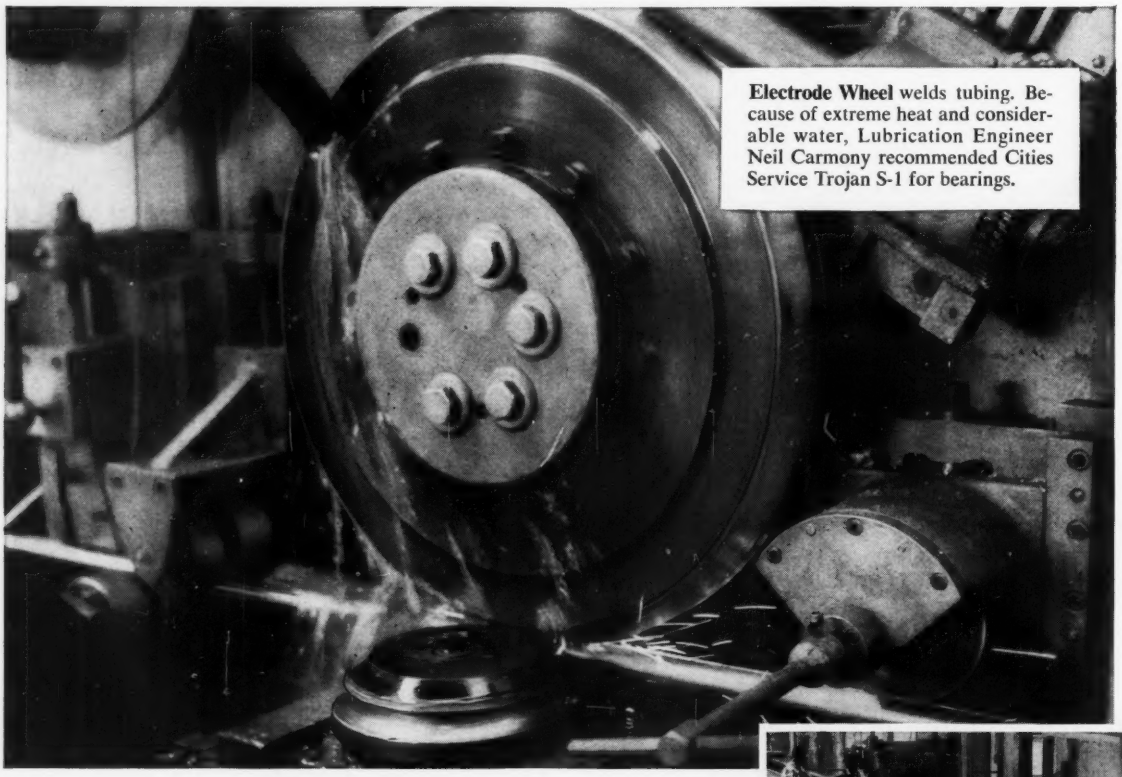
Pipe scale, core sand, paint scale and metal particles obviously are the result of inadequate cleaning.

In machine assembly, the interiors of all housings, crankcases, reservoirs and piping used for lubricant circulation should be kept clear of scale, rust and other accumulations.

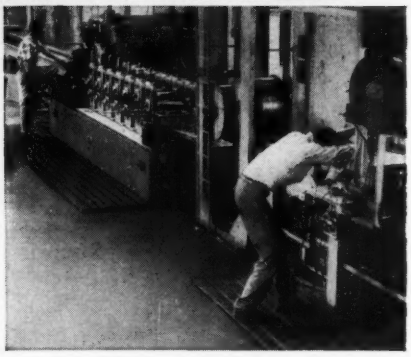
The oil reconditioning system must be able to

(*Cont. on p. 447*)

New Tubing Mill Produces at Speed of 350 Feet Per Minute!



Electrode Wheel welds tubing. Because of extreme heat and considerable water, Lubrication Engineer Neil Carmony recommended Cities Service Trojan S-1 for bearings.



It's Cities Service for Hydraulic System and all lubrication

On January 1, 1957, P&H Tube Corporation put a new mill in production at Bossier City, Louisiana—the only one of its kind in the United States.

Equipped with special drives and worm gears, this unique mill actually produces at the rate of 350 feet per minute—twice the speed of conventional tubing mills!

With speeds like this, you might expect lubrication troubles . . . but P&H Tube Corporation has none. One big reason is the careful study of each machine made by Cities Service Lubrication Engineer Neil Carmony—and the lubrication recommendations he made.

In hydraulic units, for example, Carmony recommended Pacemaker 200-T Hydraulic Oil . . . for he knew its ability to withstand the severest operating temperatures and speeds. He knew that due to its exceptionally high viscosity index and superior compounding, Pacemaker 200-T would give maximum resistance to thinning, oxidation, rust and corrosion.

The wisdom of this decision, as well as the choice of other lubricants shown in the chart at right, was demonstrated recently when P&H Tube Corporation ran off 125,000 feet of tubing in an ordinary eight hour shift—with normal time-out for change-overs!

That's production with a capital P! And if you'd like to improve your production picture, a Cities Service Lubrication Engineer can help you, too. Call the nearest office or write: Cities Service Oil Company, Sixty Wall Tower, New York 5, N. Y.

What the Cities Service Lubrication Engineer Recommended

Hydraulic units	Pacemaker 200-T
Bearings	Trojan H-2 Grease
Drive units	Pacemaker Oils
Bearings by the mill's electrodes	Trojan S-1

CITIES SERVICE

QUALITY PETROLEUM PRODUCTS

Vol. 14, No. 10

Lubrication Engineering

leading our league

"In the laboratories, the study of lubrication goes forward as engineering strives to meet the new problems of high speed, high temperature, and corrosive environment."

The foregoing introduced *Mechanical Engineering's* digest of 1957 literature on "Bearings, Lubricants, and Lubrications."

As our eyes wondered over the pages, we asked ourselves, "Who's the pacesetter in these studies? Who has actually published and presented at meetings the greatest number of papers in the field of bearings, lubricants and lubrication?"

Well, we took out our trusty abacus, and we came up with an answer that is most rewarding for our Society. The Engineering Societies' Index had recorded 208 references to literature in the field. Eliminating foreign journals and government technical reports, a total of 120 references remains. Out of those 120 cited in the digest, 44 were either published in *Lubrication Engineering* or presented and sponsored by ASLE—a percentage of 36.6. This constitutes the greatest number of references to any one group listed in the digest; in fact, it is greater than the next top three groups combined.

For an organization that has been in existence for only 14 years, this is truly an outstanding record. It is a glowing tribute to the Society, its research scientists, authors, program committees and the others who solicit articles and papers for ASLE meetings.

With the advent of ASLE *Transactions,* we should add even more to our laurels, for during 1958 we will have published almost twice as many articles as during 1957.

It's good to be at the top of the field—we predict that we'll be there for a long time to come. *C. L. Willey, Adm. Sec.*

Heavy Duty High Output Internal Combustion Engine Lubrication

*W. M. Kauffmann**

ABSTRACT

Reliable performance of heavy duty high output Diesel and gas engines depends upon design of components to reduce thermal gradient of critical parts. Lubrication problems are reduced by correct bearing load, improved engine design and heat transfer. Turbocharging is firmly established as a means of increasing output. Lubrication of turbochargers becomes an important factor in trouble free operation.

TURBOCHARGING SYSTEM

Figure 1 is a diagram of the high pressure charging system. The method of high output turbocharging used by the author's company (1) utilizes basically a rather simple and efficient means of increasing engine output by employing components which contribute to optimum operating efficiency, both from mechanical and thermal considerations. Some of the factors which contribute toward higher operating efficiency of the four cycle engine are:

1. Higher volumetric efficiency and increased air flow of cool air are achieved due to improved manifolding and use of thruflow scavenging. Smaller and therefore cooler operating valves are obtained with the four valve head. Streamlining of the valve ports results in improved flow coefficient of the engine.
2. Flood-cooled pistons with controlled oil cooling reduce ring belt temperatures to a minimum thereby increasing ring life.
3. Dry exhaust manifold permits utilization of maximum of heat energy to the gas turbine end of the turbocharger. It also serves to reduce the heat rejection load to the jacket water cooling system by elimination of cooled exhaust piping. Radiator and heat exchanger costs are subsequently reduced.
4. Fuel injection characteristics are improved by means of proper tailoring of the Diesel combustion chamber and piston contour for spray dispersion and penetration.
5. High pressure turbochargers operate to maintain high air excess and moderate charging pressures. A high combined efficiency of centrifugal blower and gas turbine elements is obtained with the axial flow turbocharger by means of specifically matched engine and turbocharger. A separate self-contained lubrication system for the turbocharger retains optimum performance and reliability.
6. Air aftercooling serves to cool the blower discharge air and thereby lower the initial cycle temperature and increase charge air density to the cylinder.
7. Valve overlap is designed to obtain optimum scavenge ratio and increased trapped air volume, with a fixed valve timing.
8. High velocity jacket water cooling contributes to reduced line wall temperature. Tests indicate that with constant heat flux and a given film coefficient of heart transfer, wall surface temperature is reduced with increased flow velocity. Thermal stresses are therefore lowered in critical parts.

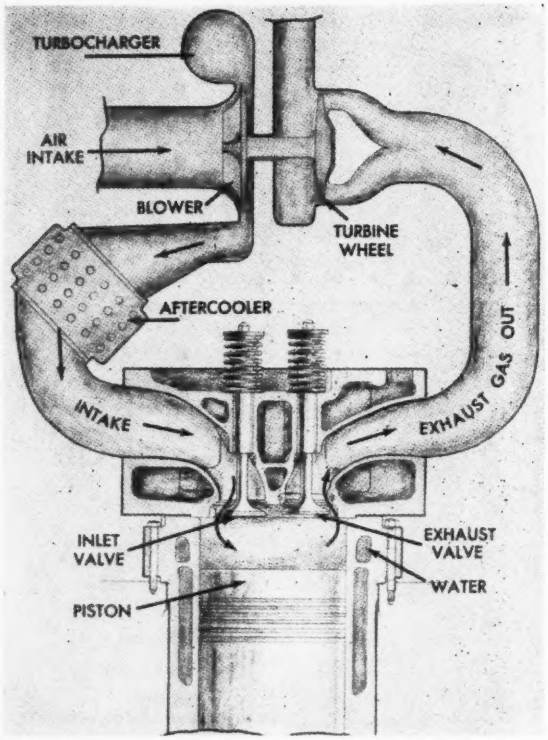


Figure 1. Diagram section of four cycle high pressure turbocharging system.

ENGINE CONSTRUCTION

Figure 2 shows the SW14, a 14 in. x 18 in. @ 450 rpm Vee Diesel Engine. High output turbocharging as utilized requires only moderately high charging pressures while at the same time maintaining a satisfactory positive differential between charging air pressure and exhaust gas pressure to turbine. This provides minimum back pressure and results in cooler operation of valves, pistons, rings, heads and cylinder liners. Combustion pressures of the SW14 Diesel at 168 BMEP

* Worthington Corp., P. O. Box 953, Buffalo, New York

rating do not exceed 1100 psi with air manifold pressure of 8.5 psi gauge. Compression pressure normally is 690 psi with a net effective compression ratio of 11 to 1. Recorded fuel economy at 110°F manifold air temperature is .359 lb. BHP/hr at full load. Favorable thermal load characteristics at rated output are indicated by a scavenge ratio of 1.65 and an air excess ratio of 1.75.

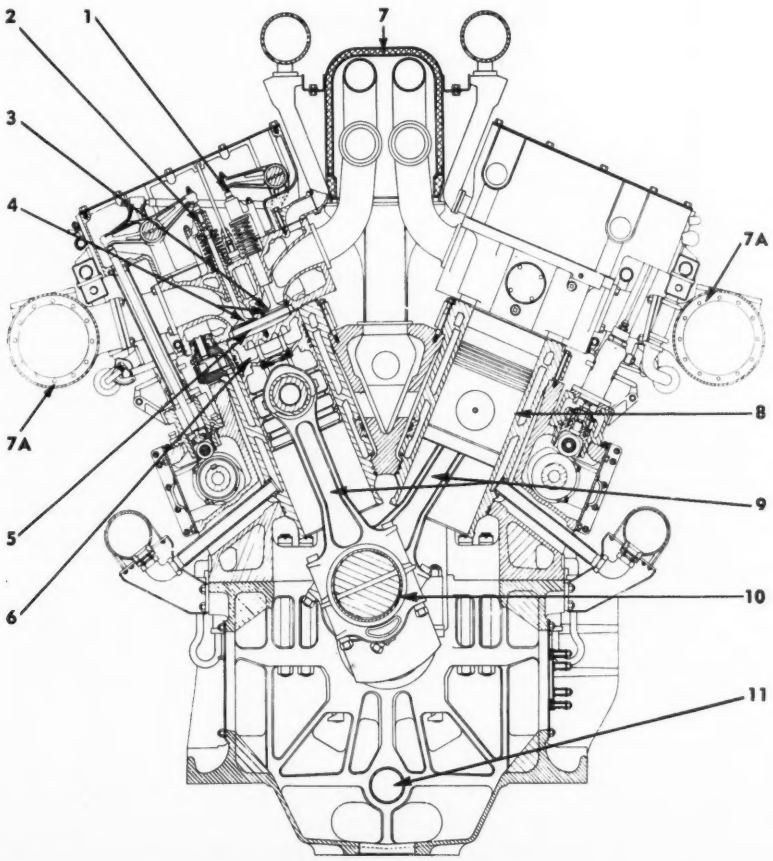
Figure 3 is a cross section of the SW-9 Diesel engine, 9 in. x 11 in. at 900 rpm. The novel air induction manifold, here referred to as jet swirl, emphasizes the significance in inlet air ram effect and turbulence that has been placed on the high pressure turbocharging combustion process. Figure 4 shows the path of induction air flow to the cylinder. The SW-9 engine has operated with clear exhaust to 225 BMEP. The combustion pressures at 168 BMEP were 1200 psi with .375 lb BHP/hr. recorded fuel economy. In this case, charging pressure was 12.3 psi gauge with a calculated air excess ratio of 1.9 indicating high flow characteristics and low cycle thermal loading.

Having reviewed the basic design and premise for the high output engine, the relation of lubrication to the problem of operating efficiency and reliability is of immediate importance. Therefore, it becomes essential to discuss, in detail, the improvements which contributed to reducing thermal gradients and assisted in preventing critical lubrication problems in these units.

The author has consistently maintained that the only insurance for a well-designed piston ring to function as expected is to provide adequate cooling of the piston itself.

PISTON DESIGN

Figure 5 shows a cross sectional view of the piston for the high pressure turbocharged engine. Since the compression rings are narrow to effectively seal the combustion gases and prevent excessive blowby, they cannot serve as a path of heat transfer from the piston to the jacket walls. Thus an effective heat dam is essential to prevent excessive thermal load at the piston ring wall, and it is equally essential that the surface under the piston crown be adequately ribbed to provide increased cooling area for the heat now dammed up in this zone. This has been termed "waffle grid" design. The cooling oil flows to the piston cavity through an eccentrically milled slot in the piston pin bore from an annular passage in the pin assembly. To improve heat transfer, the piston crown and ring belt demands lighter, thinner sections. Nodular, or ductile iron, has been adopted for the piston material because of its excellent thermal shock resistance and high strength. This new design, by improving the compression ring environment, has also demonstrated exceptional performance at high output without carboning up or sticking of piston rings due to excessive



1. VALVE LIFTERS
2. TAPPET GUIDES
3. VALVE SEAT INSERTS
4. VALVES
5. COMBUSTION CHAMBER
6. PISTONS
7. EXHAUST MANIFOLD
7A. AIR INTAKE MANIFOLDS
8. CYLINDERS
9. CONNECTING RODS
10. BEARINGS
11. LUBRICATION OIL DUCTS

Figure 2. SW14 Vee Diesel engine.

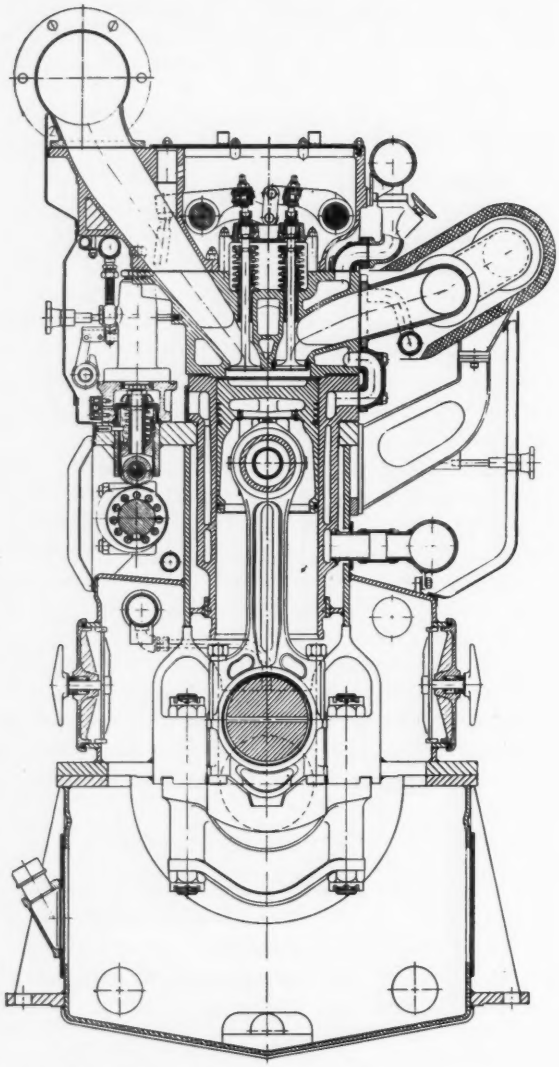


Figure 3. SW19 Diesel engine.

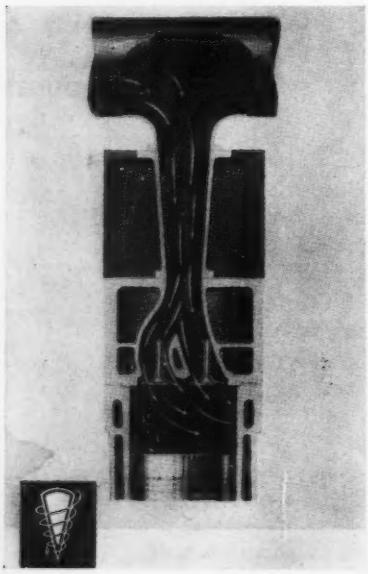


Figure 4. Aerodynamically perfect passage gives minimum flow resistance. Directional baffle imparts swirling motion to air entering cylinder for perfect combustion fuel air mixture.

thermal load. Lubrication performance has been of the order of 20,000 BHP hrs per gallon in field operation.

PISTON RINGS

Effective lubrication and minimum wear are contingent upon the piston ring configuration (2). Successful experience has confirmed the trend toward narrower compression rings and the conformable type oil rings. Top compression rings are chrome plated and lower compression rings are taper-faced to assist in rapid break-in. Four power rings are considered adequate for pistons up to 16 in. diameter, providing correct clearances are maintained.

It is essential that the piston crown be free from valve cutouts to obtain a uniform temperature patterns over its surface. This design also results in a perfectly cylindrical heat dam above the ring belt. The combustion chamber is dished, and the major volume is located in the cylinder head. Other advantages result from this execution, namely high squish air volume during combustion and improved cooling of the valve seat jacket areas in the cylinder head.

Working clearances, heat transfer, power and oil ring performance are related to the correct film lubrication of the piston. Effectively cooled pistons will function with less top clearance than a poorly cooled design. Oil rings are contained above the wrist pin.

CYLINDER LINER DESIGN

Figure 6 is a sectional view of the one-piece cast iron liner and jacker, with high velocity cooling band at the upper end directly below the cylinder head. This also provides intensive heat transfer at the walls adjacent to the top compression ring zone of the piston.

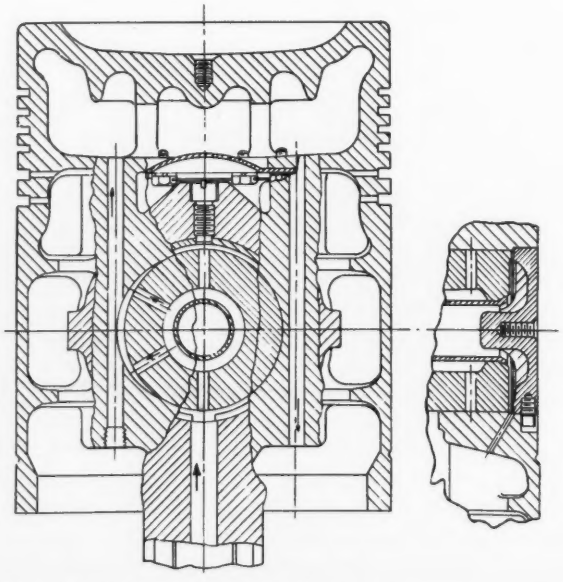


Figure 5. High output piston (section).

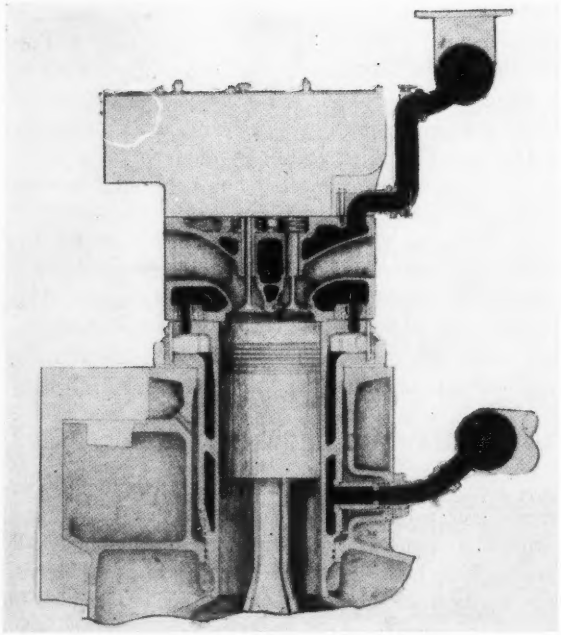


Figure 6. Cylinder liner design.

The integral liner serves two purposes. It provides rapid circulation of the jacket coolant and prevents water contamination of crankcase lubricant which has always been a potential maintenance difficulty with sleeve type liners having neoprene sealing rings. This problem becomes more critical when steam or high temperature cooling is employed as in vapor phase cooling (3).

BEARING DESIGN

Figure 7 demonstrates the size of crankshaft employed in the SW-14 Vee 16 Diesel engine. After careful analysis of the future potential output expected of this unit, a 14 inch diameter main journal shaft was designed. Thus the shaft diameter is equal to bore diameter. Bearing loads at rated load are well within conservative practice, and lubrication with babbitt-lined cast iron backed bearings has been entirely satisfactory. A standard .007 inch thickness of lead-base babbitt has proven as well adapted to this high output as it had with normally low pressure supercharged units.

Since the bearing loads are determined from the vector sum of the gas forces and inertia forces, it is essential that the maximum values of mean bearing load be conservative for the expected increase in combustion pressure. These engines potentially are capable of firing pressures of 1500 psi and up, depending on materials of construction used. Connecting rod shells have no groove in the rod half, short slots being provided at the sides to convey oil to the wristpin. The cap half is grooved to cause this flow path to be practically continuous. As oil film thickness and rate of flow are associated with bearing temperatures, clearances would appear to be of critical importance. Normal working clearance of .0005 in. diameter plus .001 inch have been found suitable for satisfactory bearing lubrication for the conditions investigated using cast iron back shells.

SURFACE FINISH AND TREATMENT

Engine manufacturers of the large heavy duty type have given importance to break-in procedure in order to prevent scuffing and scoring of vital parts. Pistons and cylinder liners which are treated to provide a .0015 inch phosphate coating have a 30 to 45 microinch finish with oil absorbent qualities. Tinnizing of piston surfaces has corrected incipient scuffing problems. Liner surfaces, after break-in, range in finish from 20-25 micro in. and provide satisfactory film retention of the lubricant.

Cam surfaces and cam rollers are likewise parcolubrized to assist in the proper run-in of these heavily loaded parts.

Wristpins on the larger engines are also being given superfinish treatment in the order of 5- to 10 micro in.

LUBRICATING OIL SYSTEM

Figure 8 represents a typical lubricating oil circulating piping layout for an SE96 500 KW Diesel engine stationary power plant. This is a relatively novel arrangement.

The engine oil drains by gravity to a large built-in base oil sump, from which it is taken by the engine driven pump thru a full-flow edge type strainer to the lubricating oil cooler through a temperature regulating valve. The cooled oil is returned to the engine through a full flow filter to the main engine oil header.

The preference for locating the full flow filter ahead of the main header is to prevent the entrance of

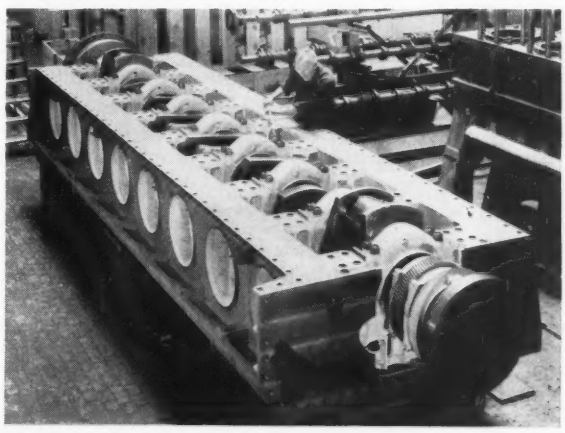


Figure 7. SW14 Vee engine base and crankshaft.

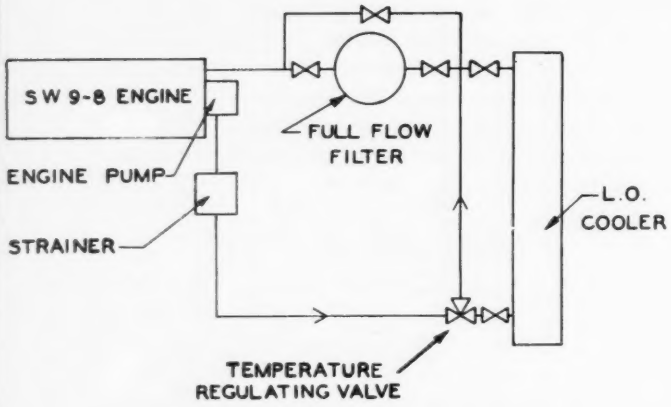


Figure 8. Full-flow lubricating filter system.

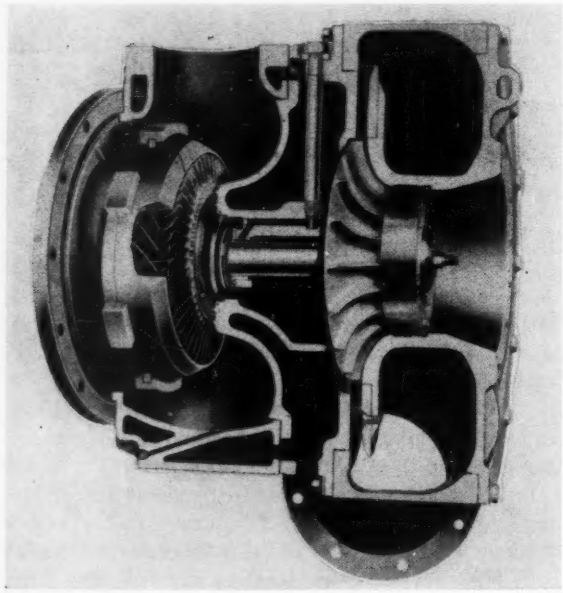


Figure 9. High pressure turbocharger.

debris and welding beads lodged in the piping and assemble into bearings during the initial run-in of the unit. It is felt that the full-flow system of filtering maintains the highest level of oil cleanliness as well as the lowest level of contaminants. Altho of higher initial cost, nominally not more than 2 percent of the initial installation, reduced maintenance and longer engine life more than repay this investment.

The author is inclined to agree that it is not so important that the filter be able to take out particles down to say 1 to 2 microns, as it is to maintain a particle size level of 8 to 10 microns. Filtration systems that will prevent 10 microns (.0004 in) contaminants from entering the engine will provide a satisfactory level of cleanliness for the high output unit.

Bypass type filters will be found inadequate for high duty service. Also centrifuges, if used, should be supplemented with full flow filters, since they will not handle the full capacity of the engine lubricating oil system. In either of the above, only a small amount, roughly 5 percent, can be handled at any time, resulting in less efficient lubrication. Aside from lower initial cost of bypass filtration, it is not considered comparable to the fulflow degree of filtration.

Engines of 2000 to 5000 BHP are generally provided with separate lubricating oil sumps, placed at the side or end of the foundation, in the basement of the power plant. Motor driven before and after lubricating oil pumps are provided to pre-lubricate the bearings of the unit prior to starting.

TURBOCHARGER LUBRICATION

The high pressure turbocharger (Figure 9), designed for pressure ratios up to 2.5 to 1 is rugged proven equipment. For large industrial engines, the gas turbine housing is water-cooled in parallel with the engine cooling system. Oil sealing is provided by means of special grooving. It is generally conceded that a separate system will permit using the best oil specifications, namely a heavy duty anti-foaming turbine oil. Therefore, an oil sump is built into the turbocharger bracket, and is provided with cooling coils for circulating water. A full-flow filter and positive displacement engine driven turbocharger lubricating oil pump are incorporated into the engine.

With the integrated turbolubricating method, only annual inspection and servicing is necessary. Thorough bearing inspection and cleaning may extend to a four-year interval, depending on the installation and maintenance procedure. Inefficient operation of the turbocharger, reflected by reduction in air pressure may be traced to dirt deposited on the impeller of the centrifugal blower. A high degree of filtration and efficient air filtering of the combustion air will contribute to the proper performance of the turbo-machinery.

The Buchi type turbocharging system is employed in the SW-14 and SW-9 engines. Therefore, the turbine blading is of the combined impulse and reaction type to utilize the blow-down energy of the exhaust gases upon opening of the exhaust valve. The pulses are transmitted to the turbine through small diameter pipes, and a number of cylinders may be combined in any one pipe. Normally two pipes enter the turbine on 6-cylinder units. These obtain the minimum interference and provide adequate pressure differential during the charging and valve overlap period. Turbine speeds are in the order of from 15,000 to 20,000 RPM for engines in a range of 1500 to 5000 BHP. At these speeds, no abnormal lubrication problem exists if proper filtration and cooling are maintained. Normal lubricating oil and water temperatures are both held at 160°F for continuous, heavy duty operation.

LUBRICATING OIL SPECIFICATIONS

It has been stated correctly (4) that the engines

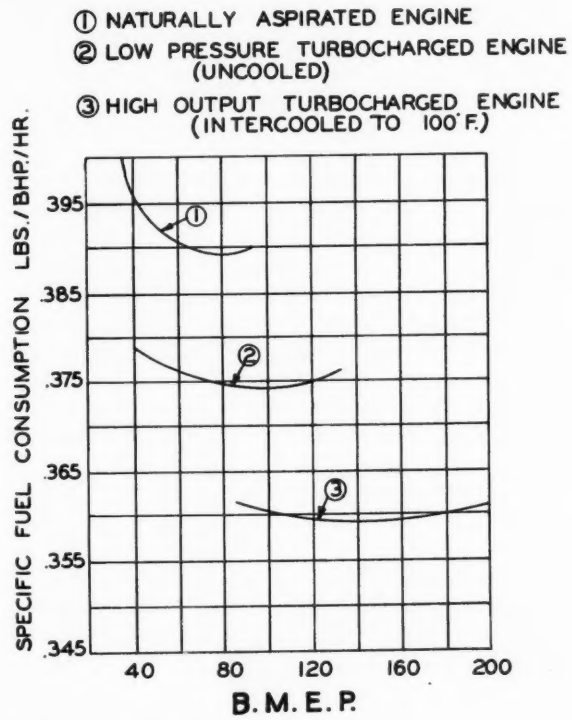


Figure 10. Typical fuel economy curves.

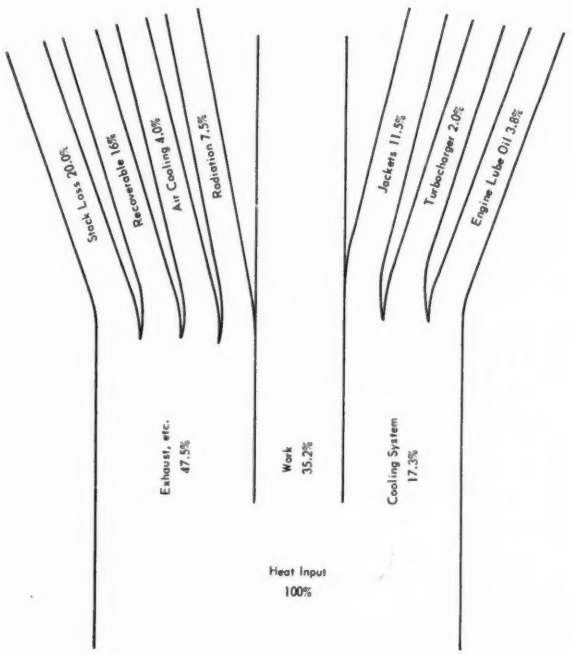


Figure 11. Heat distribution chart based on .375 lb/BPH/hr Diesel operation

manufacturer has no control of the factors which influence proper maintenance after the unit is installed. His responsibility lies in providing a design capable of operating efficiently within an "oil specification" range determined by him. The constant increase in ratings and speeds of commercial engines requires changes in lubricating oil recommendations from time to time. More emphasis is given to viscosity, resistance to heat, low carbon formation, and nonsludging characteristics, Briefly, these four properties determine the ability of an oil to distribute itself throughout the engine, its resistance to oxidation, its behavior in the upper compression ring zone toward preventing carbon and its ability to keep the engine clean by keeping contamination in suspension.

The trend toward additive type oils with these characteristics is unmistakable. The necessity for new methods of evaluating oil changes by frequent chemical analysis becomes important. More recently the use of residual fuels has demonstrated the superiority of high additive content in lubricants. An oil having a viscosity between the limits of 700 to 1000 at 100°F and 70 to 85 at 210°F having a flash point of 420-450°F should cover the requirements for practically all types of heavy duty Diesel and gas engines. Where normal crankcase temperatures and jacket temperatures exist (not over 160°F), oils nearer the lower end of this viscosity range have been found satisfactory. Where higher crankcase temperatures of 180°F and jacket temperatures of 200 to 250°F are encountered, oils near the upper limit are preferable.

The multiple functions of crankcase oils for high output modern engines have been summarized (7) as follows: to prevent wear, reduce friction, act as a cooling medium, act as a sealing agent, retard formation of deposits on engine parts, act as a hydraulic oil in the hydraulic valve lifters, prevent rust and corrosion. By this token, internal combustion engineers have become conversant with the technology of additives such as oxidation and corrosion inhibitors, detergent dispersants, rust preventives, anti-wear agents, foam inhibitors, pour-point depressants, and viscosity index improvers.

It is not the purpose of this article to delve into the manifold problems of high output heavy duty engine lubrication which may develop as a result of unusual or extreme condition of operation. Typical of these are cold weather starting, low temperature operation, high ambient temperature, steam cooling of jackets and residual fuel or inferior grades of fuel oils.

It is also generally conceded that poor Diesel combustion, smoky exhaust, improper injection setting, poorly cooled pistons, extra harsh piston ring oil control or badly worn rings cannot be corrected by change of lubricant specifications. These must be corrected and controlled before extensive deterioration of engine parts results.

HEAT BALANCE

Directly related to lubrication efficiency is combustion efficiency. This, in turn, is reflected in a reduction in heat rejection to the jacket water cooling system and lubricating oil. It is a paradox of the high out-

put unit that the mean temperature of the cycle is approximately the same as the low pressure charged engine, although power output has increased 50 percent. The heat rejection to cooling offers a criteria for mean cycle temperature. Figure 10 shows typical fuel economy curves for naturally aspirated, low pressure, and high pressure turbocharged Diesel engines.

Assuming full load ratings at 80, 120 and 170 BMEP respectively, typical fuel economics would be .390, .375, and .360 lb. per BHP/hr.

The heat balance of an engine indicates the heat rejected to the circulating water and lubricating oil and permits visualizing the thermal characteristics of the unit. Therefore, one must consider the high output engine as an internal combustion engine, and cooling means for the charging air, in order to differentiate it from other types of power.

Figure 11 represents a typical heat balance chart for this unit. The heat lost to the cooling water is 11.5% and including loss to turbocharger 13.5%, the heat loss to oil is 3.8%, or a total of 17.3%, which is the lowest loss to cooling recorded by an internal combustion engine.

Based on .360 lb/BHP/hr fuel economy, jacket heat loss would be 940 Btu/BHP. A low pressure charged Diesel having .375 lb/BHP/hr fuel consumption and a typical 20.5% heat loss would obtain 1490 Btu/BHP/hr. A 60 percent increase in output would obtain equal heat rejection on this basis. Since the heat rejected to the cooling water of an engine is an indication of the mean thermal load of the cylinder walls of the combustion space, heat stresses and temperatures of the vital parts are unmistakably related to this value.

Heat flow in a given engine increases with load, and heat transfer from the gas to the cooling medium depends to a large extent on the variation in gas temperature and heat transfer phenomena. It has been established that gas temperature during the combustion cycle, under constant conditions, varies with charge air temperature and rate of air excess. In the high output engine, due to after-cooling, charge air temperatures increase only slightly with load. Since the charging process becomes more efficient at increased load, a moderate change in mean temperature of the cycle in the higher load range is experienced. Tests reported recently (5) show liner wall temperatures adjacent to the piston top ring varying from 125C (257°F) at 120 BMEP at 155C (312°F) at 225 BMEP with uncooled aluminum pistons, or a 20 percent temperature increase for an 88 percent increase in output. These refer particularly to the MAN high pressure charged test engine research. Oil cooling is currently employed in these units. However, the relation in temperatures undoubtedly still prevails.

AFTERCOOLING AIR CHARGE

Brief comment regarding after cooling of the turbo lower discharge air will assist in the correct application of this auxiliary equipment. The limit of cooling is determined by the available water supply temperature, and this normally does not exceed 90°F. It is therefore possible to obtain a 10 degree approach to the air manifold, or 100°F to cylinders where adequate supply prevails.

Since air cooling of high output heavy duty engines is so closely related to combustion efficiency, the manufacturer must establish tenable limits for extreme operating conditions. Where unsuitable water supply exists, a closed system aquatower or evaporative cooler will permit reducing temperature to withstand five degrees of the wet bulb, provided that humidity is low at the job site. In other applications, a radiator closed system is employed. However, here it is probable that the approach to ambient, of the cooled water, will not exceed 15 degrees, so that manifold air temperature at 95°F ambient would become 120°F.

As the pressure ratio of the turboblower increases, the heat rejection to the aftercooling water will increase owing to the higher air discharge temperature. The moderate pressure ratios of the system described in this article obtain the minimum size of aftercooler for relative output and thermal loading conditions. The effect of aftercooling on thermal load of an engine is shown in Figure 12 as calculated by Birmann (6).

TWO CYCLE HIGH OUTPUT ENGINE

Unprecedented developments in the turbocharging of heavy duty two cycle engines have been reported within the last few years, both for high output oil—Diesel and spark ignition gas engines.

Figure 13 is a schematic section of a turbocharged uniflow scavenged two-cycle high output spark ignition unit. The basic components comprise a pulse-reaction type axial flow supercharger in conjunction with after-

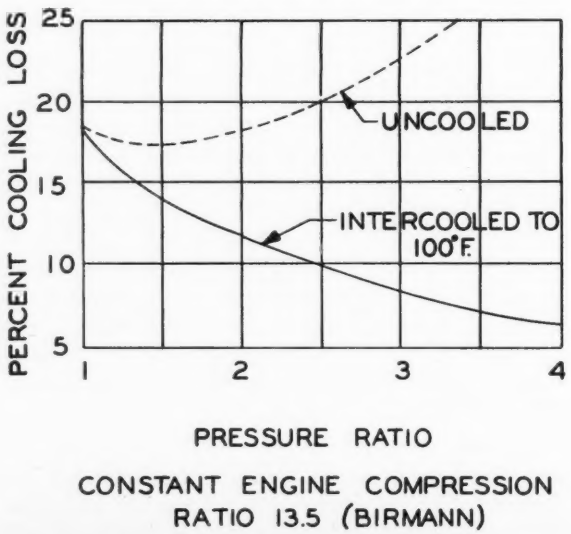


Figure 12. Effect of air cooling on heat rejection.

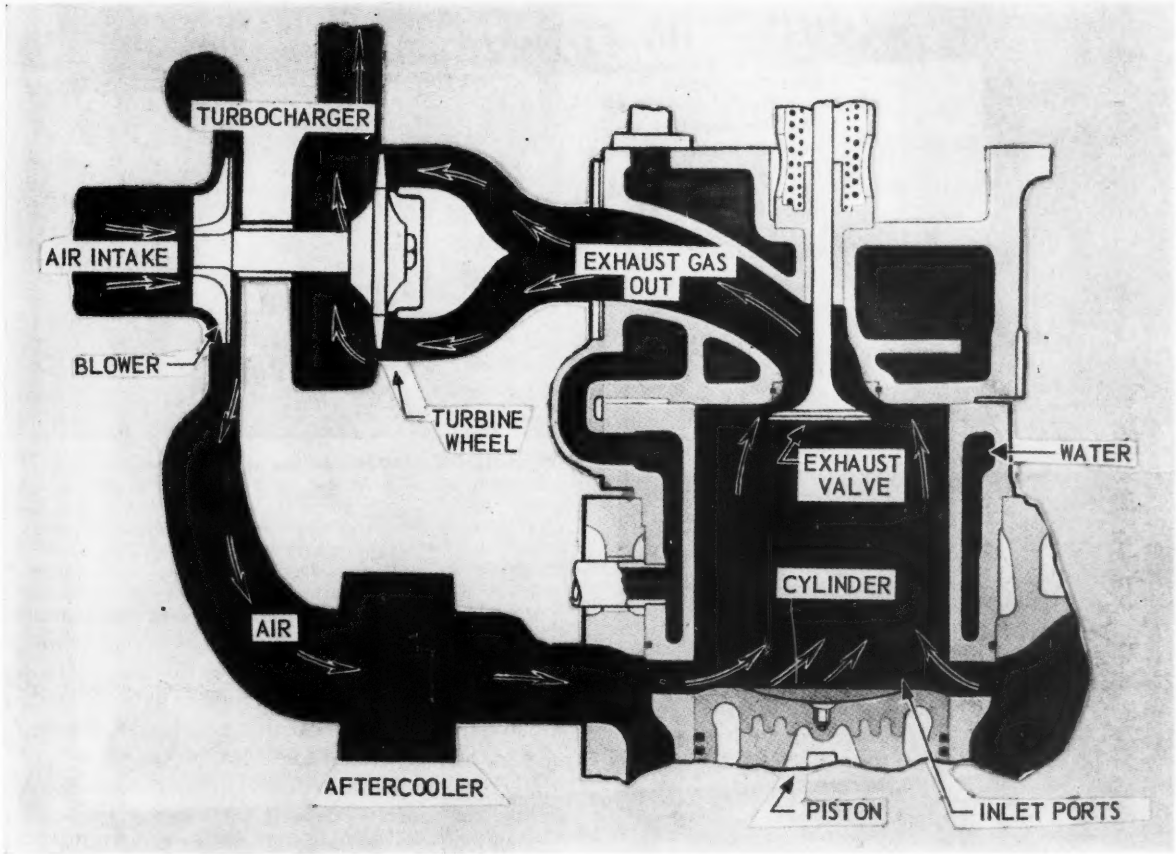


Figure 13. Uniflow two cycle turbocharging system.

cooling. Pressure ratios in this case, for ratings of 100 BMEP, average 1.50.

Where:

$$\gamma = \frac{Pa + Pb}{Pb}$$

Pa = manifold air pressure
Pb = barometric pressure
γ = pressure ratio, blower side

With intensive aftercooling and extra high pressure turbocharging, research at loads in excess of 135 BMEP and pressure ratios of 2.0 are currently being expedited. Compared with 77 BMEP of the pump-scanvenged version, a 75 percent increase in output is being realized.

Figure 14 shows a view of the piston construction of the final design. Cooling oil from the rod is conveyed through an annular space around the wrist pin carrier to a nozzle which disperses a spray of oil at the center of the crown under surface. The oil drain is maintained at a high level so that adequate cocktail shaker effect is obtained by the reciprocating movement of the piston.

Coupled with the high charge air ow during the scavenging period, the piston surface and liner walls are effectively cooled internally to produce an environment favorable to efficient lubrication in the most critical zone.

PERFORMANCE

The record of performance of the high output heavy duty internal combustion engine is no longer a matter of conjecture. A large number of units operating continuously in critical areas give ample evidence of reliability at ratings of 168 to 175 BMEP four cycle and 96 to 105 BMEP two cycle. These ratings will increase of economic necessity as research information is accumulated at the higher loads.

Representative of these installations is a municipal power plant of a midwest utility shown in Figure 15. This view shows an SW14 Vee 16 cylinder dual fuel engine nominally rated 4000 BHP at 160 BMEP. Reported lubricating oil consumption of 19000 BHP hours per gallon represents a substantial savings in lubrication costs. Reduction obtained in fuel consumption has warranted full 24 hour operation although load varies drastically over this period.

Figure 16 is typical of a small western utility installation using an SW 96 oil Diesel engine rated 750 BHP at 160 BMEP. Currently operating to furnish irrigation power these engines are employed in a wide range of service.

The two cycle turbocharged application is represented by Figure 17 which shows a petrochemical process plant utilizing a number of these units for compression of gas. The lugging power at reduced speed is

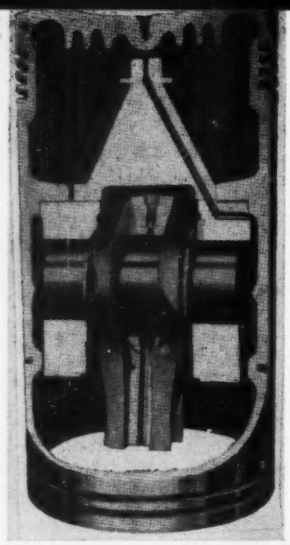


Figure 14. Two cycle high output engine. Below: Figure 15. (top) SW14 Vee dual fuel engine. Figure 16. (center) SW96 Diesel engine generating set. Figure 17. (bottom) SUTC turbocharged uniflow two cycle gas engine.

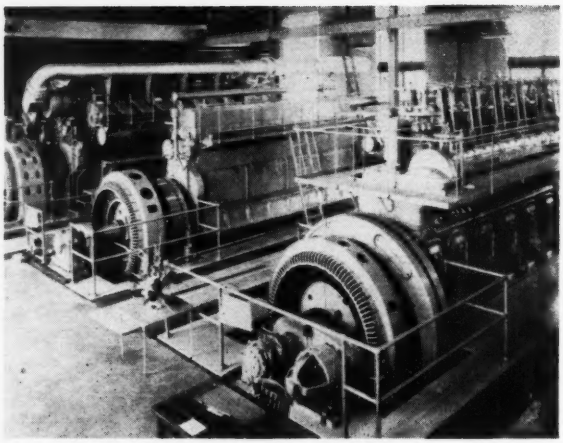
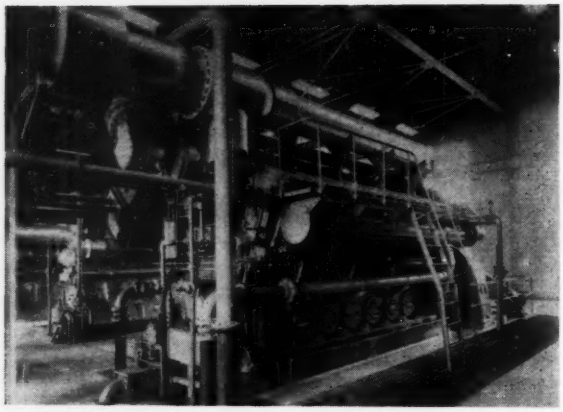
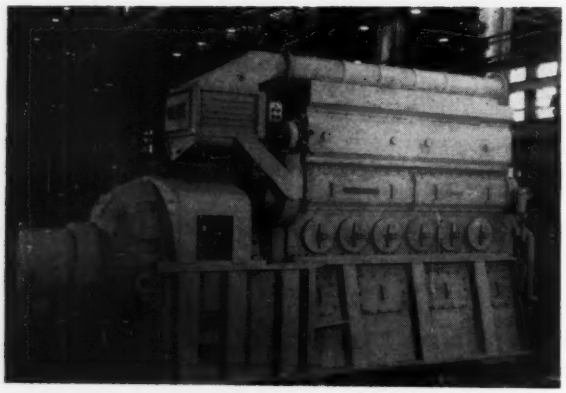
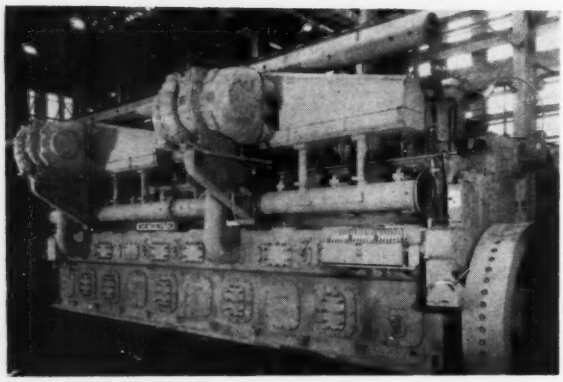


Figure 18. SW1416 dual fuel engine installation.







particularly important in this type of service as it presents a more severe lubricating problem than it would in normal operation. As outlined in this article, the design characteristics of the engine have permitted trouble-free running with no lubrication difficulties due to thermal or mechanical loading.

Figure 18 shows, more vividly than words can describe, the progress made in the internal combustion engine industry. In the forefront is a 750 BHP Diesel installed in 1930. The second unit is a 1000 BPH Diesel installed in 1940. In the rear is the SW14 Vee 16 rated 4200 BHP installed in 1955 in this midwest municipal power plant.

It can truly be said that those who have experienced this development period find considerable gratification in these achievements. However, more startling innovations still lie ahead in the use of Diesel power. And as always, the lubrication problem will be met successfully as these go into service.

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Measuring Activity of Sulfur Additives in Oils

*by Helen Sellei**

ABSTRACT

A method has been developed for quantitatively measuring the corrosiveness towards copper of active petroleum additives containing sulfur. A dilute solution of the sulfur additive is simply heated in a test tube with a copper strip at 200 to 400°F. Such reaction products as do not flake off are removed from the strip by washing with dilute potassium cyanide. The weight loss of the strip is the measure of corrosiveness. The change in weight loss with time and temperature, can be used to characterize the activity of sulfur additives.

INTRODUCTION

Extreme-pressure lubricants function by reacting with metal surfaces to form inorganic films that protect the sliding surfaces under conditions of high temperature and pressure. Chemical agents are added to perform this function. The tendency of an extreme-pressure agent to react with a metal surface is often called its "activity." Too low an activity gives insufficient film to protect the surfaces; too high an activity results in too great an attack on the surface, or excessive corrosive wear. Sulfur and sulfur compounds are the most widely used extreme pressure additives. Under identical conditions, different sulfur compounds will protect surfaces differently, as governed by the reactivity of the sulfur atoms in the molecule.

Insight into the activity of sulfur additives might permit correlation between structure and performance under service conditions. If the extent of reaction of an additive with a standard metal surface could be reliably determined, measuring it under progressive exposure times would give a measure of sulfur activity. The manner of dependence upon temperature would further characterize additive behavior.

The corrosiveness of sulfur compounds in petroleum products is conventionally measured by the ASTM copper-strip test. Color changes (1, 2, 3, 4) can distinguish minute differences in corrosive attack by almost non-corrosive products, but they cannot differentiate between active compounds, all of which give a black strip. The amounts of sulfur found on the copper in the less-corrosive regions can be determined (5, 6, 7), but the methods fail in the black-strip region because reaction products flake from the copper surfaces (8). All of the available active sulfur can be removed from an oil through use of large surpluses of copper powder (9, 10, 11), but no practical method appears to have been devised for measuring the progressive attack on copper strips of excess sulfur in petroleum products.

The chemistry and kinetics of the reaction between copper and elemental or polysulfide sulfur (12, 13), sulfur in the atmosphere (14), and dilute solutions of sulfur in organic solvents (13, 15, 16) were extensively studied as much as 30 years ago. However, the closest approach in the petroleum field is the demonstrated increase with temperature in the reactivity of sulfur compounds (11).

To implement the study of sulfur activity in additives, gear lubricants, and cutting oils, a practical method for measuring copper corrosion has been developed. Although such additives are used chiefly with iron surfaces, they behave similarly with both metals (11) and the convenience of conventional copper strips favors use of this metal. In the method, corrosion in the black-strip region is quantitatively determined by measuring the amount of copper consumed in the reaction. The reaction products adhering to the strip are removed with a suitable solvent, and the decrease in weight of the strip is determined. Additives are tested in blends resembling the finished products in which they are used.

MEASURING COPPER CORROSION

Each measurement of corrosion is carried out on a clean weighed copper strip exposed in the test sample at an elevated temperature for a predetermined time. The copper sulfide formed is dissolved from the strip with a solution of potassium cyanide, and the strip is again weighed to determine copper loss.

Copper strips 3 in. x ½ in. cut from 1/16 in. electrolytic sheet copper, are polished with No. 0 steel wool and weighed to 0.1 mg. A 15-g sample of sulfur-containing oil is placed in a 25-ml test tube 150 mm long and 16 mm in diameter. A weighed strip is immersed in the oil, and the test tube is stoppered with a cork and placed for a selected time in an oven at a selected temperature between 200 and 400F; higher temperatures could lead to irrelevant chemical

* Research Department, Standard Oil Company (Indiana), Whiting, Indiana.

changes. After exposure, the strip is washed with naphtha and chloroform, dipped in 10 per cent aqueous potassium cyanide solution for three minutes, washed with running water, dipped in acetone, dried, and weighed. Loose scale should be wiped from the strip with a clean dry cloth before the chloroform wash.

Each test condition that could affect copper loss was arbitrarily set after investigating the effect of varying it. For a typical oil containing 0.5 per cent additive sulfur, the 15-g sample contains enough sulfur to give reliably measurable copper losses. Doubling the ratio of oil to copper, by using a 30-g sample in a larger test tube, increased the copper loss only 10 per cent. Halving this ratio, by using two separated copper strips, increased copper loss 20 to 45 per cent with different samples and temperatures. These findings suggest that the geometry of the test equipment, by affecting convection in the test tube—and perhaps diffusion of sulfur to the copper surface—markedly affects copper loss. If so, the viscosity of the test sample would also be expected to affect test results.

In testing finished oils, viscosity is predetermined. In testing additives, however, it will depend mainly on the viscosity of the diluent oil used in making the test blend. As most additives tested are intended for use in gear oils, a diluent that would give a blend of essentially the viscosity of a finished gear oil would be desirable. Such a diluent is SAE 50 oil of at least 85 V.I. Although the diluent oil itself contained 0.5 per cent sulfur, copper strips immersed in it up to 16 hours at 400F lost only 1 to 4 mg; the natural sulfur content is thus too tightly bound to influence test results significantly.

The effect of diluent viscosity on copper loss was measured by testing 12 per cent blends of a commercial high-sulfur additive in mixtures of the SAE 50 oil and with hexadecane. Each blend was exposed for 16 hours at 300F Copper losses and extrapolated viscosities at 300F are plotted in Figure 1. As predicted, copper loss increases regularly as viscosity decreases. The same trend was observed with a pure sulfur compound; 1.9 per cent benzyl disulfide gave about four times as high a copper loss in hexadecane as in the SAE 50 oil.

Potassium cyanide solution selectively dissolves the reaction products without appreciably attacking the unreacted copper strips. A clean copper strip immersed in 10 per cent solution lost about 1 mg in one minute and 1.4 mg in three minutes, but only 8.8 mg in 60 minutes. Copper strips exposed to sulfur-containing oils for 2 hours at 400F were freed of scale and essentially clean in two minutes and thereafter lost about 0.2 mg per minute. The prescribed three minutes should therefore assure complete removal of products without unduly affecting the unreacted copper.

The precision of the method was checked by evaluating duplicate tests. Standard Deviation was determined as recommended for small numbers of observations (17); results calculated to % Standard Deviation were directly read from a graph (18). For 17 duplicate determinations ranging from 15 to more than 1000 mg of copper loss, Standard Deviation ranged from 0.4 to 4.8 per cent; 14 of the tests fell between 1.5 and 2.6%. Determinations by two operators on two samples in duplicate, ranging from 29 to 103 mg, gave Standard Deviations of 10.0 per cent and 5.8 per cent, respectively.

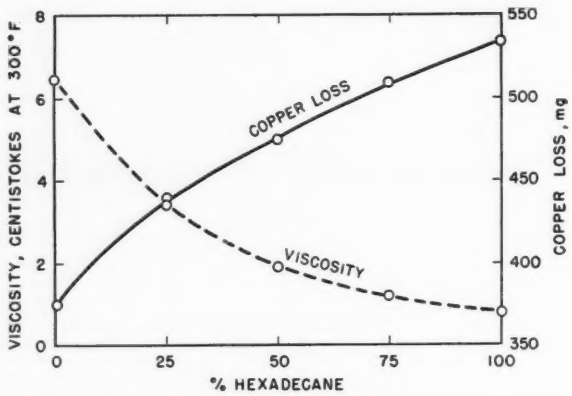

Fig. 1. Effect of Diluent Viscosity on Copper Loss

The validity of the method was checked by comparing the copper loss from the strip and the sulfur loss from the sample, and determining the composition of the scale. Of the copper lost from the strips 90-95 per cent was recovered in readily removed scale. The rest of the reaction product was determined by weight loss upon washing with aqueous potassium cyanide. The scales were analysed for copper and sulfur; the results were calculated to the amount of copper and sulfur in the total reaction product and compared to the loss of copper on the strip and loss of sulfur in the oil. Of the measured copper loss from the strips 94 to 103 per cent was accounted for. The difficulty in recovering the scales accounts for the relatively large fluctuation. Usually no copper was found in the oil; even additives known to form oil-soluble copper compounds, like sulfurized sperm oil (11), produced only negligible amounts. Of the sulfur lost from the oil, 84 to 98 per cent was indicated as recovered.

Scales from strips exposed to various sulfur additives at 350 to 400F for 16 to 72 hours were found to have an average composition of 79.3 per cent copper and 19.2 per cent sulfur. This corresponds closely to the calculated value for Cu_2S: 79.86 per cent copper and 20.14 per cent sulfur. The identity of the reaction products as Cu_2S was confirmed by X-ray diffraction; only alpha cuprous sulfide was found—no cupric sulfide or metallic copper. Cuprous sulfide deficient in copper was reported to be the product from the reaction of copper with elemental sulfur in dilute benzene solution (13).

CHARACTERIZING SULFUR ACTIVITY

With a reliable method of measuring sulfur cor-

TABLE 1—Typical Additives Studied in Blends

ADDITIVE	% SULFUR IN ADDITIVE	% ADDITIVE IN BLEND
Sulfurized Sperm Oil	11	5
Alkyl Polysulfide	33	1.4
Sulfurized Terpene	34	1.7
Benzyl Disulfide	26	1.9
Sulfochlorinated Olefin	26	2
Sulfochlorinated Olefin Polymer	8	5

rosion, one should be able to apply it to the broader problem of sulfur activity. The change of copper corrosion with time and temperature has been studied with hundreds of additive blends and finished oils. Typical additives are given in Table 1. The SAE 50 oil used as the diluent was an acid-treated Mid-Continent oil having an API gravity of 24.6 and containing a viscosity-index improver. All of these blends contained 0.4 to 0.6 per cent additive sulfur.

Time studies carried out over the range of 1 to 72 hours showed that the distinctive differences among blended additives were observed in the first 16 hours. Typical curves for three additives and a cutting oil, tested at 400F, are shown in Figure 2. All the curves show that copper losses before one hour are lower than would be expected from the subsequent contours. This behavior might be caused by lower temperatures at the start of the test, or an induction period.

The possible effects of lower initial temperature and an induction period were checked by two sets of tests at 400F on samples containing 1 per cent sulfochlorinated olefin; the results are indicated in Figure 3. In one set, new strips were inserted in the sample every two hours for a total of eight hours; although the weight loss per hour was again greater at first than later, the total loss from the four strips was less than that from one left undisturbed for the eight hours. In the second set of tests, losses from fresh strips exposed successively 4, 4, and 16 hours were compared with that from a strip undisturbed for 24 hours, which again gave somewhat higher losses. Apparently, an induction period must be overcome with each new strip, and film thickness does not determine reaction rate.

For temperature studies, a range of 200 to 400F was used. Highly active additives will give readily measurable copper losses at 200F; less-active ones need temperatures of 300F or above. Measurements at increasing temperatures with two widely different additives are shown in Figure 4. The sulfochlorinated olefin showed a lower weight loss at 400 than at 300F. This behavior was observed with but few highly active additives; perhaps part of the sulfur reacts with the diluent oil, rather than the copper, or small amounts of volatile sulfur compounds are formed. Reduction of cuprous sulfide to copper by hydrocarbons was eleminated in the X-ray examination of reaction product.

The effects of additive concentration have also

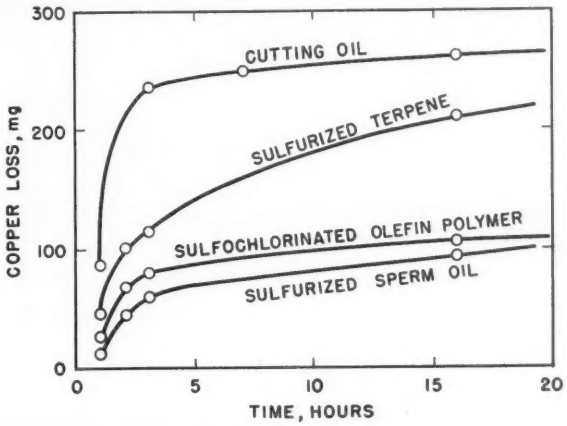

Fig. 2. Effect of Time on Corrosion at 400°F

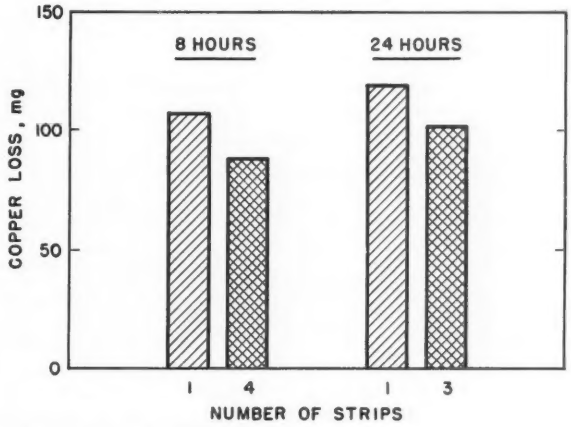

Fig. 3. Effect of Changing Strips

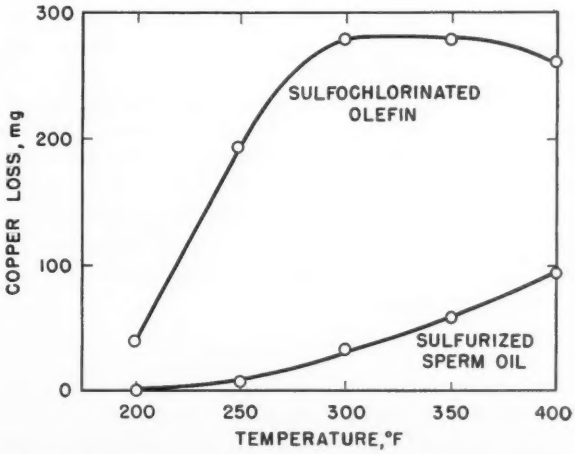

Fig. 4. Effect of Temperature on Corrosion

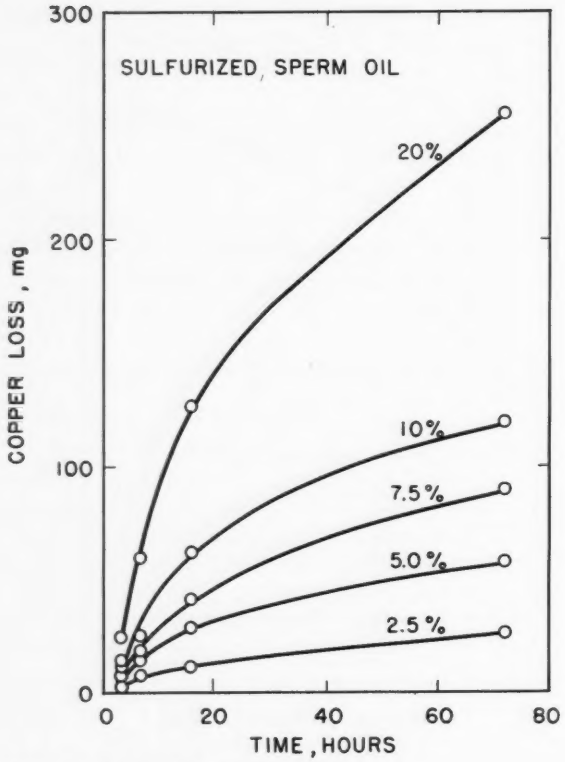

Fig. 5. Effect of Additive Concentration

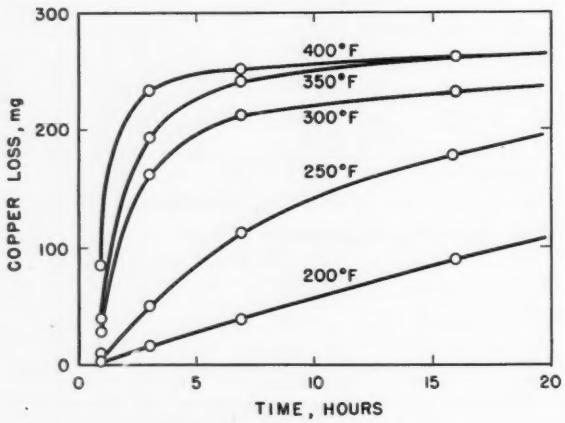

Fig. 6. Activity Pattern for a Cutting Oil

been investigated. Figure 5 shows the family of curves obtained at 300F with five different concentrations of a sulfurized sperm oil containing 11 per cent sulfur. At short times, the individual curves depend upon the activity of the sulfur present; at longer times, they asymptotically approach the amount of sulfur present. In comparing additives, they must obviously be tested at like concentrations.

CONCLUSION

A combination of time and temperature data for an additive or finished oil gives a good picture of the sulfur activity. Figure 6 shows such a plot for a cutting oil containing 0.42 per cent additive sulfur. At a fixed concentration, differences in copper-loss rate with temperature during the first few hours may well be characteristic of the specific composition tested. Future experiments will attempt to determine the extent to which such curves can be used to characterize the behavior of extreme-pressure agents in practical applications. Similar studies might also prove useful in the development of sulfurized motor-oil additives and correlating them with performance in service.

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Slope Index—An Expression for Viscosity-Temperature Characteristics

by E. E. Klaus, R. E. Hersh*, and M. J. Pohorilla†*

ABSTRACT

The Viscosity Index method for indicating the change in the viscosity of a lubricant with temperature has several disadvantages. The present paper describes a system for presenting viscosity-temperature data which has been devised to remedy these disadvantages.

Many systems have been proposed for representing the viscosity-temperature characteristics of liquids, particularly lubricating oils. In general, however, their applicability has been restricted to limited temperature ranges or to relatively narrow viscosity ranges. The Viscosity Index employed by the petroleum industry (1) is an example of such a restricted system, and the disadvantages of this method are well known (2, 3).

MORE UNIVERSAL EXPRESSION FOR VISCOSITY-TEMPERATURE CHARACTERISTICS NEEDED

Hydraulic fluid and lubricant developments during and since World War II have placed more emphasis on synthetic and polymer-thickened mineral oils which differ substantially in viscosity-temperature characteristics from the conventional oil lubricants used originally to establish the 0 and 100 Viscosity Index scale. Examples of these new fluid types include: aircraft hydraulic fluids; synthetic jet-engine oils; artillery recoil oils; shipboard gear transmission oils; non-inflammable hydraulic fluids, lubricants, and coolants; water-base fluids and lubricants; viscous damping fluids; automatic transmission fluids, and multiple viscosity-graded crankcase lubricants.

The use of many of these fluids, particularly those with military and/or aircraft applications, covers a wide temperature range. For example, hydraulic fluids and lubricants for military aircraft are used over a total temperature range of −65 to +700F. Viscosity Index, as well as most viscosity-temperature expressions are based on measured viscosities at only 210 and 100F. The current needs for expressing the viscosity-temperature characteristics for all fluids and lubricants over wide temperature ranges can be achieved only by a major modification of the V.I. system, or a completely different approach.

The current ASTM Viscosity Index system has appeared in the ASTM Standards on Petroleum Products since 1940 and was officially adopted in 1942. This method is essentially the same as the Dean and Davis Viscosity Index system which was first proposed in 1929 (4). The Viscosity Index system has enjoyed wide acceptance and usage particularly in the field of automotive and industrial lubricants of the mineral oil type.

At the present time, however, revisions of the V.I. system are being undertaken by the ASTM to correct some of the anomalies in the high V.I. region. In the development of a more universal system, therefore, it appears desirable to avoid revised scales for expressing viscosity-temperature characteristics of all fluids and lubricants which might be confused by name or value with the existing V.I. scale. A system that is simple to employ is highly desirable.

Slope Index is proposed as a universal expression for viscosity-temperature characteristics for all liquids and lubricants. The term Slope Index and its numerical values will not easily be confused with the current V.I. values. The system is relatively easy to employ and has wide applications to various fluids.

Derivation of Slope Index. The Slope Index value is based on the ASTM viscosity-temperature chart first prepared as a tentative ASTM method in 1932. The ASTM chart is based on an equation developed by Walther (5) (6) (7) (8). The form of the equation used for the ASTM chart can be represented as follows:

$$\log \log (v + k) = \log B - m \log T \qquad [1]$$

or

$$\log (v + k) = \frac{B}{T^m} \qquad [2]$$

where: v = kinematic viscosity in centistokes
k = constant (0.6 for viscosities exceeding 1.5 centistokes)
T = absolute temperature, degrees Rankine (459.7 + t F.)
m and B = constants for a particular oil

* Petroleum Refining Laboratory, The Pennsylvania State University, University Park, Pennsylvania.
† Kendall Refining Company, Bradford, Pennsylvania.

The equation is empirical and has not yet been derived precisely from any physical properties of liquids. The viscosities of a large majority of hydrocarbons and mineral oils can be portrayed as straight lines on this ASTM chart. The ASTM charts have been extended to cover the temperature range of −100 to +700F. using the Walther equation (19). A large number of synthetic fluids and polymer-thickened oils have also been applied to this chart with good results.

Table 1 summarizes the high and low temperature viscosity values of several classes of fluids. The data in Table 2 indicate both positive and negative deviations from a straight line extrapolated from 100 and 210 F. viscosities on the ASTM chart at both high and low temperatures.

Hydrocarbons and mineral oils show good general conformity to the chart at all temperatures above the cloud point of the oils (9, 10).

Dense center molecules in the synthetic fluid field including phthalate, pentaerythritol, phosphate, and carbonate esters show a low temperature curvature on the ASTM chart to give lower than predicted viscosity values.

Waxy mineral oils, polymeric fluids, and polymer thickened blends show the opposite low temperature curvature resulting in higher than the predicted viscosity. In the high temperature range (500 to 700 F.), the mineral oils show good general conformance to a straight line function on the ASTM chart.

Silicones and organic esters generally show a deviation from a straight line resulting in lower than predicted high temperature viscosities. A chlorinated aromatic hydrocarbon and tricresyl phosphate are shown on the table as illustrations of fluid which give higher than predicted high temperature viscosities.

Thus Table 2 illustrates the straight line function of mineral oils on the ASTM chart along with some positive and negative deviations from the chart at both high and low temperatures. These data indicate the good general relationships expressed by the ASTM equation and chart. The type of deviations noted cannot be improved appreciably by changes in the basic equation due to the positive and negative nature of the deviations. These deviations, however, point up the need for a versatile expression for viscosity-temperature characteristics to handle any desired temperature range for all liquids.

Several methods of expressing the slope of a line on the ASTM chart have been proposed. One of the methods offering maximum significance combined with simplicity of determination is the ASTM Slope (sometimes called Penn State Slope No.). This ASTM Slope value which has been used in the published literature since about 1940 (11, 12, 13, 14) is defined as the tangent of the angle alpha shown in Fig. 1.

$$\text{ASTM Slope} = \text{Tan}\,\alpha = \frac{Y \text{ cm.}}{X \text{ cm.}} \qquad [3]$$

where: α = the angle of the viscosity-temperature line with lines of constant viscosity on the ASTM chart.

Y = distance in cm. between the two desired viscosity values (vertical) on the ASTM chart.

X = distance in cm. between the two desired temperature values (horizontal) on the ASTM chart.

TABLE 1 Viscosities of Various Fluids

Fluid	Measured viscosity (centistokes) at −65°F.	−40°F.	0°F.	32°F.	100°F.	210°F.	350°F.	490°F.	707°F.	Viscosity index
Mineral Oils:										
Naphthenic 200 Neutral					44.6	5.74		0.87	0.45	66
Naphthenic Bright Stock			156000		201	12.0	2.61	1.16	0.57	21
Neutral-Resin Blend					61.5	9.75	3.02			135
Synthetic Fluids:										
Di-2-Ethylhexyl Sebacate	8030	1420	193		12.7	3.34	1.37	0.77		155
Methyl Silicone		290			37.5	15.2				174
Chlorinated Silicone		934	287		59.9	21.2		4.52	2.1	162
Chlorinated Silicone DC		2870	241		16.2	4.5		1.00	0.52	202
Chlorinated Aromatic Hydrocarbon					42.5	3.17		0.55		−604
Fluorinated Hydrocarbon A					5.48	1.32				−130
Fluorinated Hydrocarbon B					69.0	4.93				−229
Pentaerythritol Ester		5310	509		22.4	4.54	1.57	0.83	0.42	136
Dicapryl Phthalate		31800			26.8	4.12				24
Tricresyl Phosphate					38.3	4.48		0.76		−55
Tri-n-Amyl Phosphate		114			3.79	1.36				91
Di-n-Hexyl Carbonate		60			2.93	1.17				109
Polymer-Containing Fluids:										
Spec. MIL-O-5606 Hydraulic Fluid	2110	483	100	44	14.2	5.11	2.37			226
API Oil No. 103 (PRL 2913)		7880	655	182	33.0	8.03	3.06			168
API Oil No. 104 (PRL 2914)		5020	538		33.2	8.65				172
PRL 2924 (Polybutene Blend)		10700	817	208	33.3	7.34	2.74			161
Spec. 51-F-21 Hydraulic Gear Oil		1050	209		28.1	9.96				184

TABLE 2 COMPARISON OF EXPERIMENTAL VISCOSITIES WITH THOSE EXTRAPOLATED FROM THE 100 F. AND 210 F. VISCOSITIES FOR THE FLUIDS GIVEN IN TABLE 1 (ASTM Slope and Eby Tables (16) used for extrapolations; this is equivalent to using an ASTM Viscosity Temperature Chart)

FLUID	ASTM SLOPE 100–210 F.	PER CENT DEVIATION OF MEASURED VISCOSITY FROM EXTRAPOLATED VISCOSITY AT −65 F.	−40 F.	0 F.	32 F.	350 F.	490 F.	707 F.
Mineral Oils:								
Naphthenic 200 Neutral	0.800						−7	−13
Naphthenic Bright Stock	0.815			17		1	−4	−7
Neutral-Resin Blend	0.627					1		
Synthetic Fluids:								
Di-2-ethylhexyl Sebacate	0.701	7	8	7		−3	−9	
Methyl Silicone	0.304		0					
Chlorinated Silicone	0.315		43	5			−14	−29
Chlorinated Silicone DC	0.606		215	38			−12	−23
Chlorinated Aromatic Hydrocarbon	1.150						14	
Fluorinated Hydrocarbon A	1.115							
Fluorinated Hydrocarbon B	1.001							
Pentaerythritol Ester	0.717		−28	−15		−6	−13	−24
Dicapryl Phthalate	0.836		−56					
Tricresyl Phosphate	0.895						3	
Tri-n-amyl Phosphate	0.863		−36					
Di-n-hexyl Carbonate	0.856		−25					
Polymer-containing Fluids:								
Spec. MIL-O-5606 Hydraulic Fluid	0.482	283	111	29	10	−1		
API Oil No. 103 (PRL 2913)	0.539		251	61	23	1		
API Oil No. 104 (PRL 2914)	0.505		228	61				
PRL 2924 (Polybutene Blend)	0.586		177	49	18	3		
Spec. 51-F-21 Hydraulic Gear Oil	0.390		180	48				

The resultant slopes are all negative. The minus sign has been omitted in this paper since the use of the ASTM Slope without the negative sign is customary. The ASTM Slope is in most cases a fraction (0.100 to 1.400, approx.), and improvements in a fluid's viscosity-temperature characteristics result in a *lower* number for the slope value. The ASTM Slope is definitely related to the Slope m in the ASTM equation [Eq. 1]; in the viscosity region above 1.5 centistokes this relation is as follows:

$$m = 5.056 \text{ (ASTM Slope)} \qquad [4]$$

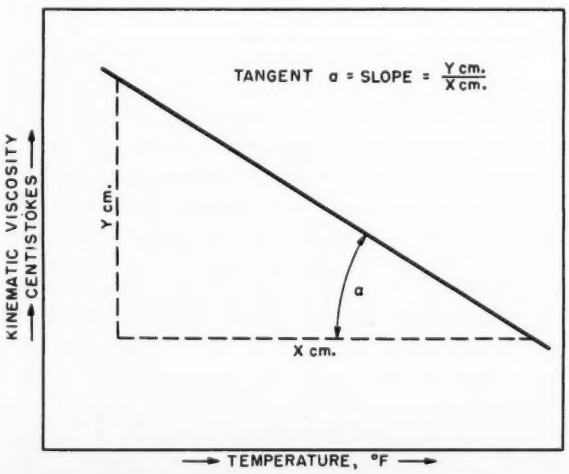

Figure 1. Method of calculating ASTM slope.

The method of calculating the slope from direct measurements on a ASTM Viscosity-Temperature chart was selected because of its simplicity over computing values of log log $(v + k)$.

While it appears logical that low values for the ASTM Slope refer to a low change in viscosity with temperature, there is some tradition and certain advantages in the association of high index values with low rate of change of viscosity with temperature. Slope Index is derived directly from ASTM Slope in a manner to remedy some of these shortcomings of the parent ASTM Slope expression. Slope Index is defined by the following equation:

$$\text{Slope Index (S.I.)} = (1.5 - \text{ASTM Slope}) \times 1000 \qquad [5]$$

The limits for Slope Index were chosen to encompass all of the liquids and lubricants currently known to be in the development or commercial stages. The Slope Index values of typical fluids are shown on Fig. 2. V.I. values from 0 to 120 are covered by the range of about 600 to 1000 units on the Slope Index scale. Esters of certain organic acids, silicate esters, and silicones cover the range of 1000 to 1300 on the S.I. scale. Aromatic hydrocarbons, fluorocarbons, chlorocarbons, and bromocarbons generally range from 50 to 600 in S.I. These compounds have negative V.I. values. A S.I. value of 1500 represents the perfect oil (equivalent to zero ASTM Slope). This liquid would exhibit the same viscosity at all temperatures.

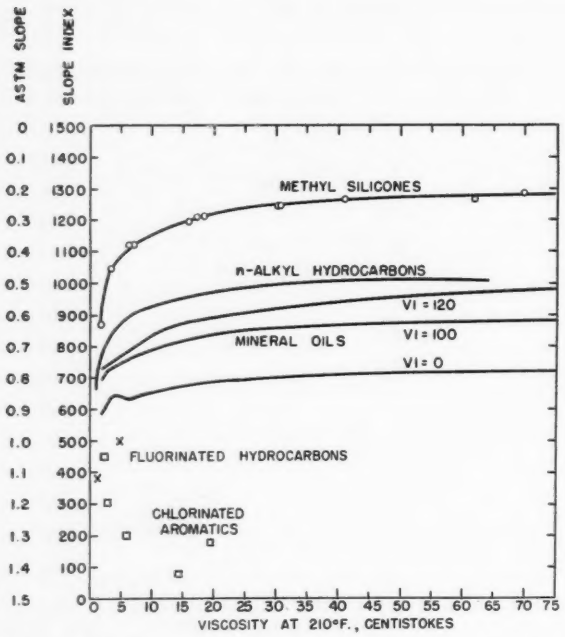

Figure 2. Viscosity-temperature characteristics of various fluids.

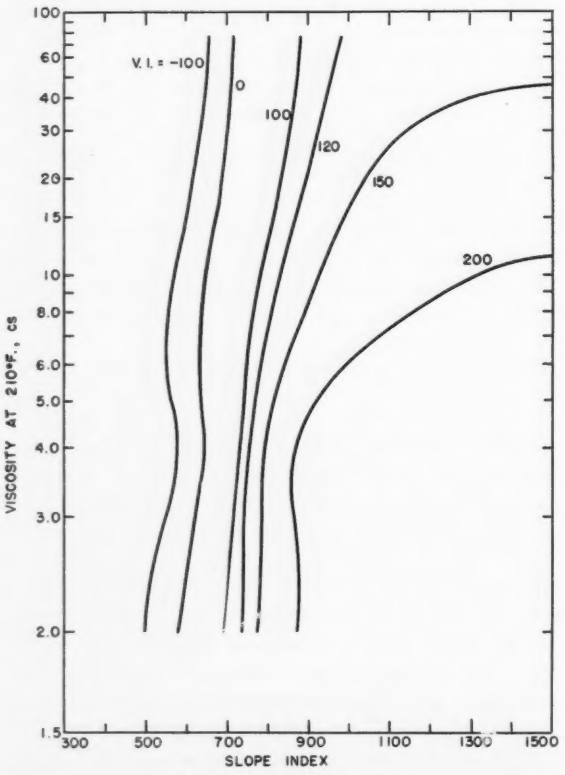

Figure 3. Comparison of slope index and ASTM Viscosity Index.

COMPARISON OF SLOPE INDEX WITH VISCOSITY INDEX

A comparison of Slope Index and Viscosity Index as a function of viscosity level is shown on Fig. 3. The relationship between Slope Index and Viscosity Index, is fairly consistent over the range of 0 to 120 Viscosity Index. This relationship, however, shows wide variations for Viscosity Index values of 200 and higher. Another evaluation of Viscosity Index and Slope Index can be made by comparing each system with the percentage change in viscosity per unit of temperature change for a fluid.

The equation for determining a fluid's per cent viscosity change per degree Fahrenheit, which is derived from the general equation for the ASTM chart, is given below (15):

$$100\left(\frac{dt}{v\,dt}\right)_t = -230.2585\,\frac{(v+k)\,\log\,(v+k)}{vT} \cdot\left[\frac{\log\,\log\,(v_1+k) - \log\,\log\,(v_2+k)}{\log\,T_2 - \log\,T_1}\right] \qquad [6]$$

where v, k, and T have the same significance as in Equations [2].

The percentage change in viscosity per degree Fahrenheit is a property of a liquid that can be measured directly. It is a simple and informative one. The variation of this percentage change with both temperature and viscosity levels limits the use of this expression, per se, to define viscosity-temperature properties.

Comparisons of Viscosity Index and Slope Index with percentage viscosity change per unit of temperature change are shown on Figs. 4 and 5, respectively. It can be seen that Slope Index is a regular or smooth function of both viscosity level and per cent viscosity change over the entire range of values compared. Where Viscosity Index values are 150 and higher, the percentage viscosity change per unit of temperature change is not a consistent function with increasing viscosity level.

A plot of log log (centistoke viscosity + 0.6) for the 100 F. versus the 210 F. viscosity values has been used to illustrate various schemes for expressing the viscosity-temperature characteristics. Such a plot for Slope Index and ASTM Slope is shown in Fig. 6. The portion of the V.I. scale suggested for retention is also shown on this figure. The lines of constant Slope Index on this type of plot are straight and parallel. Geniesse (17) compared eight viscosity-temperature proposals in this fashion. The Slope Index function and one other method produce straight lines, and only the former method results in parallel lines. The portion of the V.I. scale shown on Fig. 6 illustrates the relatively small part of the overall scope of the Slope Index System that is currently covered adequately by the Viscosity Index system.

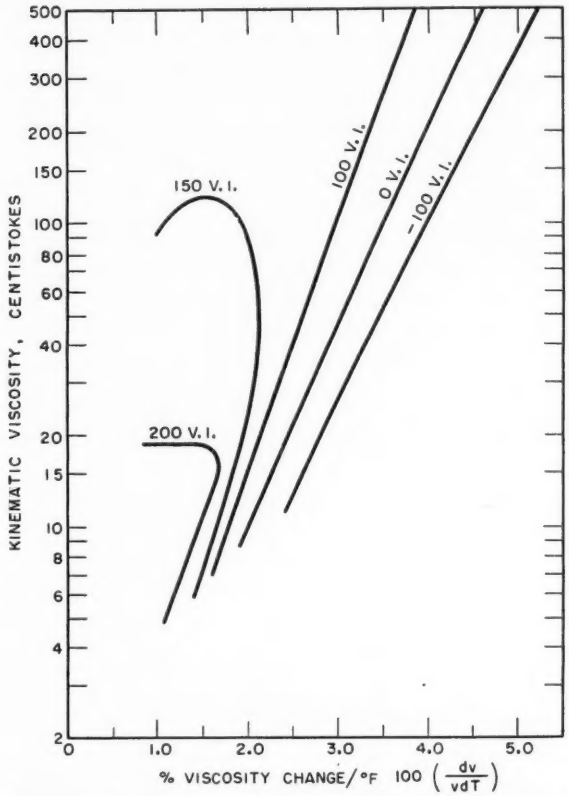

Figure. 4. Comparison of ASTM Viscosity Index and rate of change in viscosity with temperature at 100°F.

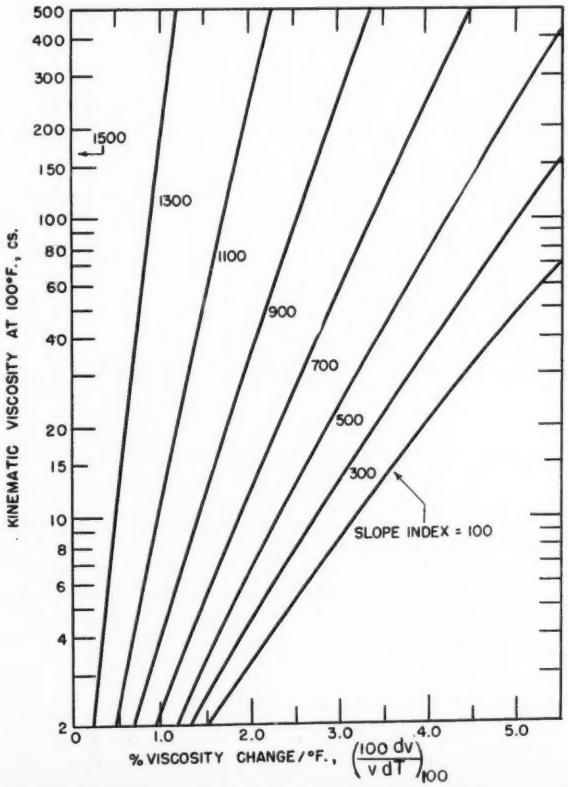

Figure 5. Comparison of slope index and rate of change in viscosity with temperature at 100°F.

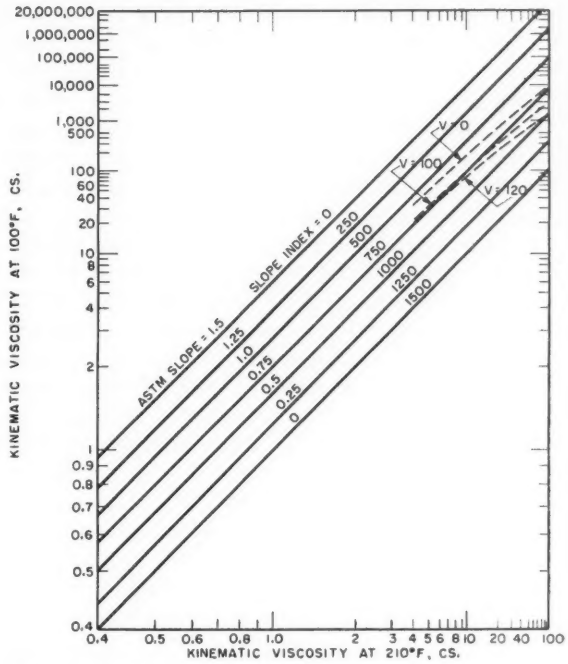

Figure 6. Lines of constant slope index.

ILLUSTRATIONS OF APPLICATION AND SCOPE OF SLOPE INDEX

Perhaps the outstanding advantage of Slope Index is the extreme simplicity of obtaining the S.I. value. Values accurate to within 5 units of Slope Index can easily be determined with an ASTM chart and a ruler for measuring the X and Y distances involved.

The same overall accuracy can be obtained from a Slope Index protractor of the type shown on Fig. 7. This instrument is used by placing the straight edge indicated by the arrows along the viscosity-temperature line of the test fluid on the ASTM chart. The reference point is placed on any horizontal (constant viscosity) line. The slope is read from the point of intersection of this same horizontal line with the grad-

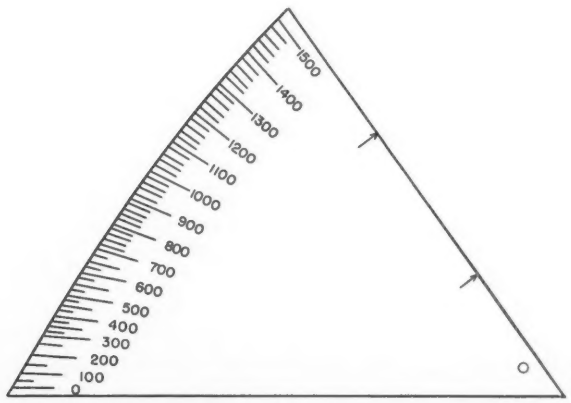

Figure 7. Slope Index Protractor.

uated edge of the protractor. The Slope Index protractor is prepared by substituting the Slope Index value corresponding to the angle on a standard protractor scale.

Tables have been prepared by Eby (16) for the determination of the ASTM Slope value used in the Slope Index equation. Using these tables to determine Slope Index, values accurate to one unit of Slope Index can be attained. The Eby tables are based on more precise measurement of distances on the viscosity and temperature scales of the ASTM chart. The ASTM Slope used to calculate Slope Index can be obtained from these tables by subtracting the slope scale value at 210 F. from the slope scale value at 100 F. A simple calculation is required to use the Eby tables to determine Slope Index for any temperature range from −100 to +700 F. The versatility of using Slope Index over any desired temperature range is one of the unique features of this system that appears particularly desirable in view of the present trends toward measured viscosities over wider temperature intervals for almost all fluid and lubricant applications.

The Slope Index system employing simple measurements on the ASTM Chart or simple calculations from the Eby tables also provides for: (A) the extrapolation and interpolation of viscosities over the wide temperature range defined by the chart; (B) the determination of the temperature at which a fluid possesses a particular viscosity; and (C) a method for obtaining the composition of simple binary blends to meet a required viscosity level.

The use of Slope Index as a parameter in preparing blending curves for polymer solutions in various base stocks is illustrated in Table 3 and Fig. 8. It should be emphasized that most of the blends represented are in the range of Viscosity Index well above 120 where erratic behavior limits the usefulness as well as obscures the meaning of a similar blending curve based on Viscosity Index.

The family of curves shown in Fig. 8 includes the base stocks and finished fluid compositions for several commercial functional fluids including two Navy hydraulic gear oils, an Army recoil oil, an aircraft hydraulic fluid, and a preservative fluid for aircraft hydraulic systems. The six base stocks and the lines of constant polymer concentration show the regularity of the Slope Index function and the convenience of this type of curve in obtaining blends of desired viscosity levels at any temperature as well as establishing the viscosity-temperature characteristic of the blend.

The additivity of Slope Index in simple binary blends of mineral oils and synthetic fluids is demonstrated on Table 4. These data show again that Slope Index is useful for a wide range of fluid types and is not limited by fluid type or viscosity-temperature characteristics. Data for ASTM Viscosity Index are also shown on this table for comparison.

The use of Slope Index over wide temperature intervals with various fluid types is demonstrated on Table 5. Intervals ranging from the conventional 100 to 210 F. as well as −65 to +700 F. are shown. The Slope Index for a straight line on the ASTM chart will be the same for any temperature interval. Therefore,

TABLE 3
TYPICAL BLENDING DATA FOR POLYMETHACRYLATE AC 58 IN MINERAL OILS

MINERAL OIL BASE STOCKS*	POLYMER CONCN., WT. %	KINEMATIC 210°F.	VISCOSITY (CS.) AT 100°F.	SLOPE INDEX
A	0	1.48	3.74	734
	2	2.10	5.03	887
	4	2.92	7.00	973
	6	4.05	9.85	1034
	8	5.53	13.7	1076
	10	7.38	18.7	1108
	12	9.38	24.2	1132
B	0	1.66	4.61	723
	2	2.33	6.04	874
	6	4.52	11.9	1019
	8	6.22	16.7	1063
	12	10.3	29.0	1113
C	0	1.87	5.79	708
	4	3.66	10.4	946
	8	6.94	20.2	1051
	10	9.02	26.8	1081
	12	11.2	33.9	1101
D	0	2.08	7.11	696
	6	5.63	17.4	994
	8	7.65	24.2	1037
	12	12.1	39.9	1086
E	0	2.26	8.37	686
	4	4.62	15.8	920
	8	8.62	30.1	1022
	10	11.2	40.2	1051
	12	13.3	48.4	1069
F	0	2.50	10.3	675
	2	3.82	15.2	816
	6	7.48	29.6	960
	8	10.1	40.8	1003
	12	14.8	61.2	1049

* Base Stocks B, C, D, and E have been prepared by blending mineral oils A and F.

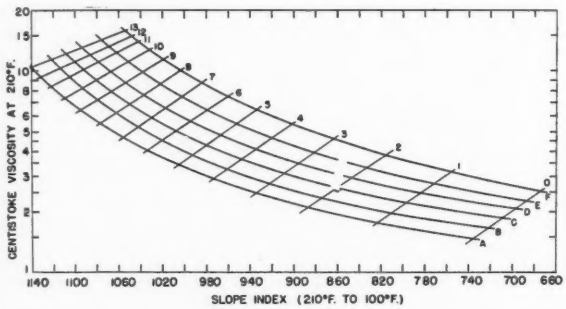

Figure 8. Use of slope index in a blending chart for polymethacrylate AC 58 in mineral oils.
Numbers indicate per cent polymer concentration; lines adjacent to numbers indicate constant polymer concentration. Letters indicate base stock properties; base stocks B, C, D, and E prepared by blending mineral oils A and F.

deviations from linearity at high and low temperatures can easily be detected quantitatively by calculating the Slope Index for various temperature intervals based on measured viscosities. Simple methods have been described for the measurement of viscosities in modified Ostwald viscometers over the −65 to +700 F. temperature range.

There has been some recent interest in the ASTM method for extending measured viscosities for automotive lubricants from the current range of 100 to 210 F., e.g., from 0 to 350 F. The effect of this change in temperature interval is illustrated in Table 5 for non-waxy mineral oils, waxy mineral oils, and polymer thickened mineral oils.

Summary of Advantages of the Slope Index. Slope index is based on the ASTM viscosity-temperature chart and the Walther equation which are widely used since their publication by the ASTM in 1932 and adoption in 1939. Plots of Slope Index versus percentage viscosity change per unit of temperature change, and lines of constant Slope Index on plots of log log viscosity (centistokes +0.6) at 210 F. versus 100 F., show the regularity of the Slope Index function with these more fundamental viscosity-temperature parameters.

TABLE 4 COMPARISON OF MEASURED AND CALCULATED SLOPE INDEX VALUES FOR BLENDS OF VARIOUS FLUIDS

SYSTEM	COMPOSITION (VOL. %)	VISCOSITY, CS. 100°F.	210°F.	V.I.	SLOPE INDEX MEAS.	CALCD.	% Dev. FROM MEAS.
Mineral Oils (18):							
EF	100E	392.1	16.31	−5	676		
	50E50F	94.06	7.985	32	674	665	−1.33
	100F	31.92	4.419	7	653		
BJ	100B	221.2	18.08	97	826		
	50B50J	80.90	9.735	108	802	796	−0.75
	100J	34.21	5.612	113	766		
LC	100L	100.3	10.86	100	797		
	50L50C	53.97	7.206	101	765	764	−0.13
	100C	30.09	4.914	92	731		
ND	100N	116.7	9.639	53	710		
	50N50D	48.59	5.744	45	676	680	+0.59
	100D	21.45	3.583	14	649		
JQ	100J	34.21	5.612	113	766		
	50J50Q	21.44	3.978	83	718	697	−2.92
	100Q	13.45	2.719	13	628		
Pa. Heavy Oil		76,240	633.8	103	890		
Pa. Neutral		11.94	2.817	85	704		
50-50 Blend A-2		332.0	26.02	108	871	797	−8.50
Coastal Neut.		22.50	3.666	11	649		
Pa. 140 Bright Stock		438.3	29.61	102	862		
50-50 Blend B-2		95.14	10.15	95	781	756	−3.20
Mid-C Neut.		21.66	4.105	97	734		
Pa. 140 Bright Stock		438.3	29.61	102	862		
50-50 Blend C-2		87.67	10.54	110	816	798	−2.20
Blends of various fluids:							
Silicone A		43.40	17.86	170	1213		
Silicone B		55.80	19.52	164	1175		
50-50 (Wt.) Blend		47.94	18.34	167	1195	1194	−0.09
Di-2-Ethylhexyl Sebacate		12.59	3.305	153	797		
Silicone B		55.80	19.52	164	1175		
50-50 (Wt.) Blend		24.45	7.34	184	1013	986	−2.66
Di-2-Ethylhexyl Sebacate		12.59	3.305	153	797		
Pentaerythritol Ester		60.90	8.947	125	838		
50-50 (Wt.) Blend		23.60	4.963	150	818	818	0
Naphthenic Mineral Oil		44.62	5.742	66	700		
Pentaerythritol Ester		60.90	8.947	125	838		
50-50 (Wt.) Blend		46.34	6.574	102	761	769	+1.05
Di-2-Ethylhexyl Sebacate		12.59	3.305	153	797		
Dichloroxylenehexafluoride		1.060	0.508	—	400		
50-50 (Wt.) Blend		5.098	1.619	75	640	599	−6.40

TABLE 5 CALCULATED SLOPE INDEX VALUES FOR VARIOUS FLUIDS OVER DIFFERENT TEMPERATURE INTERVALS

FLUID	VISCOSITY INDEX	SLOPE INDEX FOR TEMPERATURE INTERVAL INDICATED							
		100-210F.	−65-100 F.	−40-210F.	0-210F.	0-350F.	0-490F.	0-707F.	100-707F.
Mineral Oils:									
Naphthenic 200 Neutral	66	700							631
Naphthenic Bright Stock	21	685			685	687	673	662	655
Neutral-Resin Blend	135	873							
Synthetic Fluids:									
Di-2-ethylhexyl Sebacate	155	799	794	794	792	785	764		
Methyl Silicone	174	1196		1196					
Chlorinated Silicone	162	1185		1162	1180		1161	1130	1117
Chlorinated Silicone DC	202	894		828	862		839	795	785
Chlorinated Aromatic Hydrocarbon	−604	350							
Fluorinated Hydrocarbon A	−130	385							
Fluorinated Hydrocarbon B	−229	499							
Pentaerythritol Ester	136	783		799	797	771	745	679	644
Dicapryl Phthalate	24	664		696					
Tricresyl Phosphate	− 55	605							
Tri-n-amyl Phosphate	91	637		676					
Di-n-hexyl Carbonate	109	644		673					
Polymer-containing Fluids:									
Spec. MIL-O-5606 Hydraulic Fluid	226	1018	909		988	995			
API Oil No. 103 (PRL 2913)	168	961			921	937			
API Oil No. 104 (PRL 2914)	172	995			954				
PRL 2924 (Polybutene Blend)	161	914			882	900			
Spec. 51-F-21 Hydraulic Gear Oil	184	1110			1069				

Slope Index covers the entire viscosity range for all liquids and can be used with equal ease over any temperature interval over the range of −100 to +700 F. This single system, therefore, covers all practical applications relating to liquids and lubricants including crankcase lubricants, jet engine lubes, gear oils, hydraulic fluids, automatic transmission fluids, damping fluids, coolants, as well as many other functional fluids. It is not only possible but practical with this system to set up for each group of fluids the optimum temperature interval over which to express the Slope Index that is of most value for a particular use or application.

The scale chosen for Slope Index cannot be confused with the existing Viscosity Index scale. The Slope Index scale is designed so that a lower viscosity change with temperature is denoted by increasing numerical values of Slope Index. It has an adequate scale range so that 1 unit of ASTM Viscosity Index (between 0 and 100) is equivalent to approximately 1.6 units of Slope Index.

Slope Index provides a simple and precise technique for extrapolating viscosities to any temperature. The use of Slope Index over several temperature ranges provides a method of quantitatively determining deviations from the straight line on the ASTM chart. Polymer blending as well as the viscosity-temperature characteristics of various fluid types can be adequately predicted using Slope Index as a parameter. The use of Slope Index in viscosity extrapolation and interpolation, polymer blending, and fluid blending can readily be simplified by the use of prepared tables.

A comparison of Slope Index with ASTM Viscosity Index shows that the following specific shortcomings of the latter system are remedied and have no counterpart in the Slope Index system.

1. The Slope Index system is not hampered by the choice of actual or hypothetical reference blends or oils such as those upon which V.I. is based.
2. The Slope Index system covers all ranges of viscosity and temperature including low viscosity fluids which are not adequately covered in the V.I. system.
3. The Slope Index system avoids negative values. The V.I. system shows a greater lack of consistency in the negative region than it does in the 0 to 120 V.I. region.

The Slope Index provides a fundamental method for defining viscosity-temperature relationships which would be acceptable for both scientific and commercial use. It would not interfere with the retention of the present V.I. system in applications where the latter has been traditionally employed with success. Adoption of Slope Index would have a strong precedent in the present practice of employing both Saybolt and kinematic viscosity according to both commercial and scientific requirements.

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Lube Lines (*Cont. from p. 423*)

remove water and accumulated foreign matter. If it is of insufficient capacity the circulation of impurities can lead to build-up at pipe fittings or in restricted parts of the lubricating system. Sludging, which is often a counterpart of oxidation, will promote concentration of the resultant deposits.

The *chemical effect* of gases which may enter a lubricating system by windage or breathing will depend upon the type of gas and its acidity. Corrosive gases drawn into a compressor with the air stream can lead to surface deterioration and ultimately flaking off of solid particles which may accumulate elsewhere in the machine to interfere with lubrication.

Oxidation of lubricants can be one of the most aggravating causes of deposits. Air, water and elevated temperature, along with the catalytic effect of copper or brass, etc. have an accelerating effect. Modern oxidation resistant oils and greases do an excellent job in retarding this tendency and in keeping lubricating systems free from resultant deposits.

Stability of lubricants is one of the most important keys to deposit-free service, provided the other factors, as indicated, are realized and accounted for.

WHAT TO DO ABOUT ALL THIS

Granted that deposits are objectionable it becomes the responsibility of the lubrication engineer to study the cause and correct the conditions wherever possible.

As this can probably increase the costs of lubrication and maintenance, it becomes the responsibility of management to accept these costs with graceful appreciation. They can well be regarded as operating insurance, an assurance that lubrication can be maintained most dependably as an adjunct to unimpeded production.

GUIDE TO AVAILABLE PUBLICATIONS

INDUSTRIAL DERMATOSES—PROTECTION PREVENTION AND TREATMENT

Medical practitioners, chemists and lubrication engineers examine use and handling of cutting oils, lubricants, and solvents and discuss protective measures in terms of ointments, housekeeping, and prevention of bacterial decomposition. $1.00. Foreign $2.00.

INTERPRETING SERVICE DAMAGE IN ROLLING TYPE BEARINGS

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LUBRICATION ENGINEERING DECENNIAL INDEX

Subject and Author Indexes list papers published in *Lubrication Engineering,* Volumes 1 through 10 (1945 through 1954). $.50. Foreign $1.00

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PRODUCT LITERATURE

FLANGE-MATCHING TABLES

The familiar oil field supply store problem of what forged steel flange mates with what plug valve is solved at a glance by Bulletin T T 880, published by Tube Turns Division of Chemetron Corporation, Louisville 1, Ky., (formerly National Cylinder Gas Company). Forged Steel flanges, rated according to intended service pressure at elevated temperatures, and lubricated plug valves, usually given WOG ratings, are properly matched in the new bulletin, eliminating the need for a user to refer to several catalogs to make the proper selection. Pertinent bolting and gasket data is also provided.

REMOTE CONTROL CATALOG

Crown Controls Div. of Trimount Instrument Co. has recently made available an eight page catalog on their line of Crown hydra-trol remote controls. The catalog describes the operation and construction of these controls together with typical applications in the industrial, marine, research, and military fields. Crown hydratrol remote controls are manual-hydraulic positioning devices and feature a two direction positive stroke combined with immediate synchronization in either direction. The hydraulic system is self-contained and operation is smooth with no lag or lost motion. Full descriptions and specifications are given in the catalog concerning both the MAR and MAT type controls. The Standard MAR type remote control has a rating of 1000 pounds force in both directions while the Standard MAT type control has a rating of 200 pounds of force in both directions. Copies of the remote control catalog can be obtained by writing Crown Controls Div. of Trimount Instrument Co., 3119 W. Lake St , Chicago 12, Ill.

PROVING RINGS

Proving ring specifications and accessories are listed in a new four-page, two color bulletin. General and dimensional specifications are given for all standard capacity proving rings in both the compression and compression-tension types. Each accessory is described in light of how it facilitates use of the proving ring. Copies of Bulletin 159 may be obtained from Morehouse Machine Co., 1742 Sixth Ave., York, Pa.

SUB-MICRON FILTRATION

The Alsop Engineering Corp. of 58 Aster St., Milldale, Conn. has just released the first complete brochure dealing with sub-micron filtration and its contribution to cost reduction in metalworking. The brochure titled "12 Avenues to Metalworking Cost Reduction" covers 12 areas of cost reduction made possible by the Alsop method of submicron filtration. Also described is the filtration equipment recommended to solve the 12 metalworking problems that result from improper or no filtration. Copies of the "12 Avenues" brochure are available, on request, from the Alsop Engineering Corp., 58 Aster St., Milldale, Conn.

FLUSH VALVE DATA

A new one-piece, bottom outlet flush valve designed to reduce maintenance and minimize product build up is described in Data Sheet No. 42, now available from The Pfaudler Co., a division of Pfaudler Permutit Inc., Rochester, N. Y. The new valve features one-piece glassed head and stem, with glass-filled Teflon seat to provide maximum protection against corrosion. Glassed surface is resistant to all acids (except hydrofluoric) even at elevated temperatures and pressures and is resistant to alkalies at moderate temperatures. Smooth, one-piece interior glassed surface discourages product adherence. Pfaudler flush vlaves can be installed on existing vessels with 125 pound or 150 pound series flanges. Copies of Data Sheet No. 42 may be obtained by writing to The Pfaudler Co., 1067 West Ave., Rochester, N. Y.

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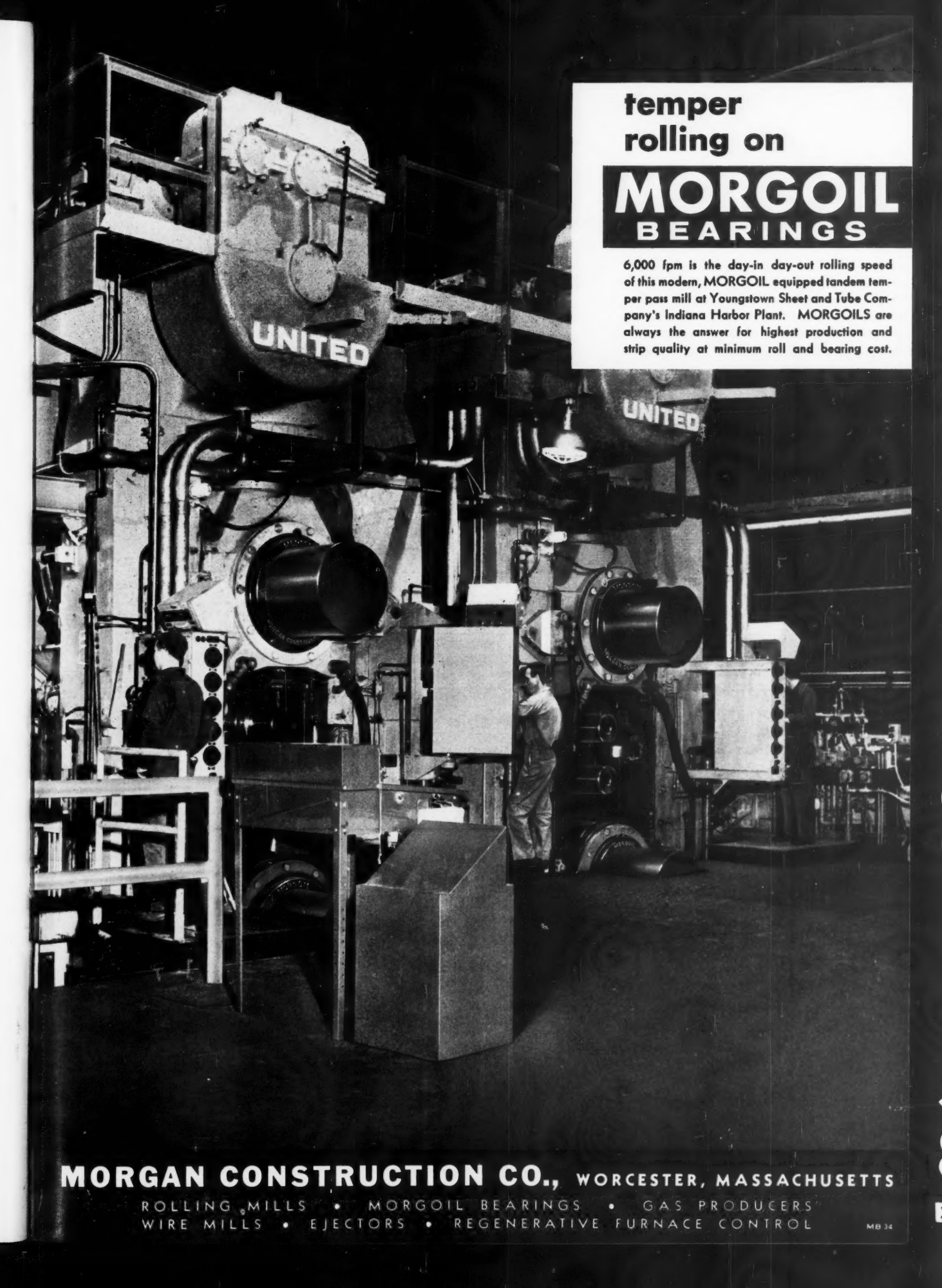

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ASLE Technical Digest

Papers which, because of space limitations, cannot be published in full in Lubrication Engineering or ASLE Transactons.

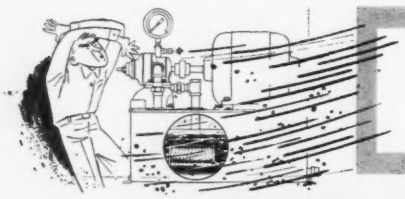
"Lubrication in a Glass Container Manufacturing Plant" by D. E. Grube

Modernization of the lubrication procedure in the glass industry today takes full advantage of available ways and means available of protecting necessary precision mechanisms against undue wear in the presence of extreme heat and dusty atmospheric conditions. Glass making requires temperatures as high as 2700 to 2800° F in furnace melting of raw materials which include sand, soda ash and lime, and cullet (crushed waste glass). Batch feeding of these materials, however, provides for utilization of the latter to protect feeder bearings against directly radiated heat; this eliminates the need for bearing cooling or usage of special high temperature lubricants. Subsequently in the container manufacturing plant the temperature may be lower, but molten glass as required for molding, even after setting and prior to annealing in the lehr, still ranges from 850 to 1000° F. Here, on mold equipment mechanisms as on conveyor bearings, the lubricant must be of adequate viscosity to withstand the temperature-reducing effect.

Effectual sealing also is provided in the design of the bearings on modern glass container manufacturing machinery in order to preclude entry of abrasive dust. A favorable consequence of this design enables the use of premium grade ball and roller bearing greases or high quality lubricating oils according to the means of application and the function of the machine. Naturally, operating conditions are considered. For example, the type of gear reducer dictates the viscosity range of the gear oil, or the need for a compounded or EP product. Fans, motors and other mechanisms in turn require consideration of the windage factor. Air flow into fans could cause oil to be drawn out of bearings, or windage in motors could cause lubricant contamination. In either case, starved lubrication, unbalanced fan rotors (due to oil collection on fan blades) or fouled windings could result.

While the intensity of operating conditions requires a considerable number of lubricants, i.e. from 15 to 17, simplified lubrication has become practicable through coordinated study of these conditions with respect to design of the mechanisms and application of automatic or centralized lubrication. The modern glass container manufacturing plant further assists the lubrication program by designating lubricants by number to facilitate ordering, handling, utilization and recommendation for new equipment throughout the plant. (*Digested by A. F. Brewer*)

Note: Unabridged copies of the foregoing papers are not available at Society headquarters. Those wishing additional information about these papers should contact the author.



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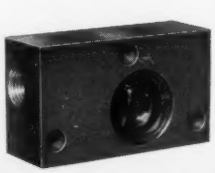

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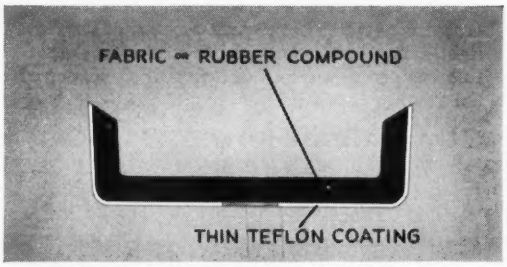

Cutaway shows how TEFLON coating is applied to outer surface of molded cup by special Garlock process. Coating becomes integral part of outer surface.

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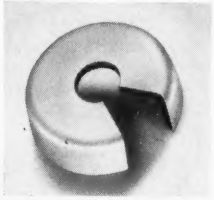

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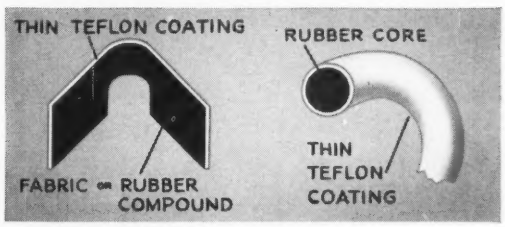

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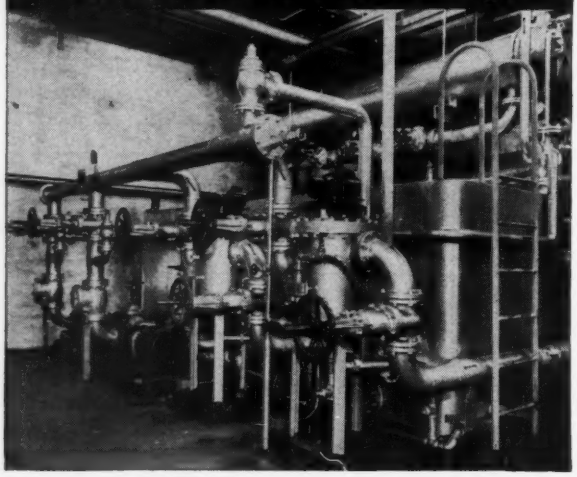

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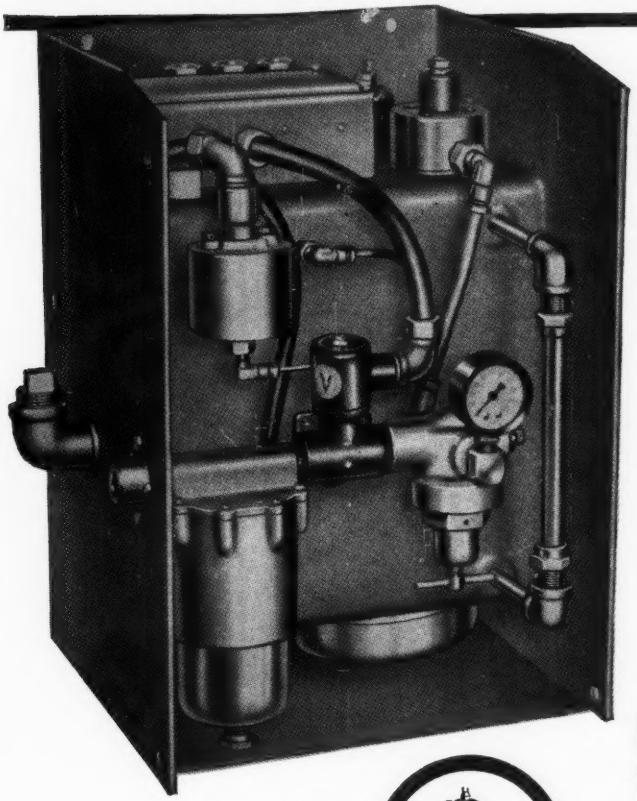

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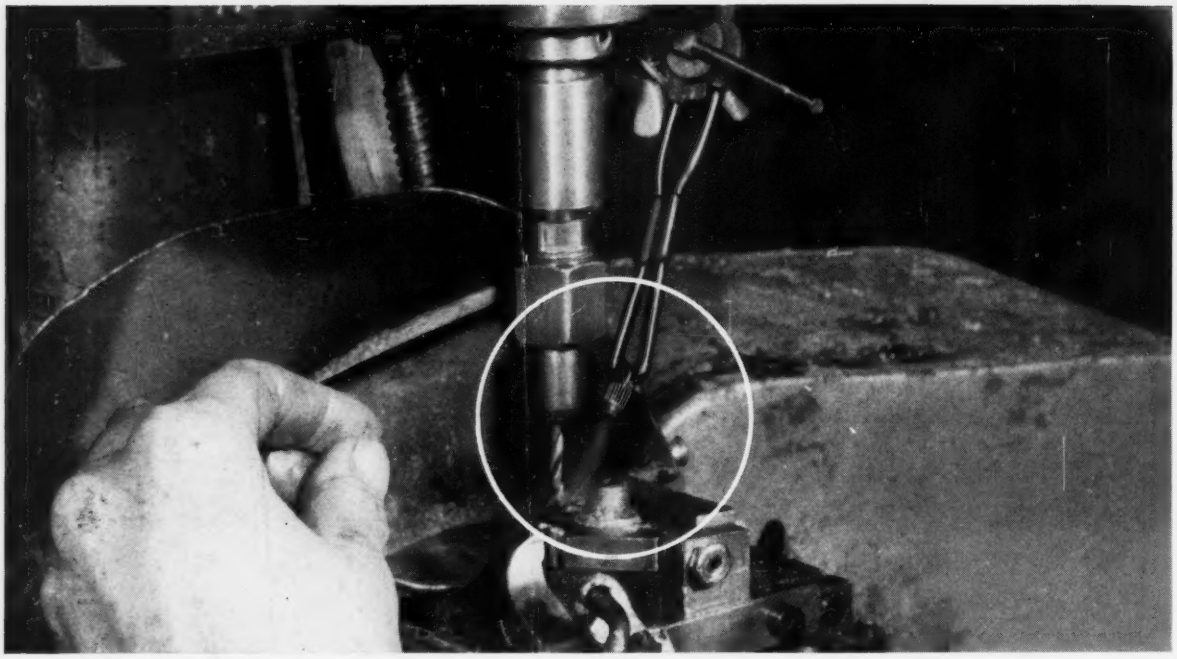

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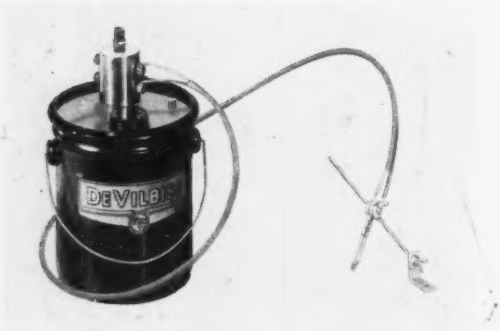

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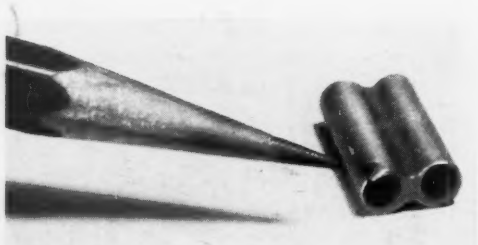

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